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Lunar Vertical-Shaft Mining System

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Executive Summary

As part of man's continuing effort to explore and expand the frontiers of space, it is believed that one day man will establish a permanent outpost on the lunar surface. Once established, significant long-term research can be conducted to better understand the lunar environment and the environment of space. To aid in this research, this report proposes a method that will allow lunar vertical-shaft mining. Lunar mining allows the exploitation of mineral resources imbedded within the surface.

The proposed lunar vertical-shaft mining system is comprised of five subsystems: structure, materials handling, drilling, mining, and planning. The structure provides support for the exploration and mining equipment in the lunar environment. The materials handling subsystem moves mined material outside the structure and mining and drilling equipment inside the structure. The drilling process bores into the surface for the purpose of collecting soil samples, inserting transducer probes, or locating ore deposits. Once the ore deposits are discovered and pinpointed, mining operations bring the ore to the surface. The final subsystem is planning, which involves the construction of the mining structure.

The structure chosen for the mining system is a geodesic-dome-shaped design based on the parabola $x^2/5$. It has a height of 45 meters and diameter of 30 meters at the base. The dome is made of ten evenly spaced rings, 5 meters vertically apart, with the tenth ring having a radius of 1 meter. Each ring will be constructed of 14 equal length members. At each member connection there are two identical support members forming the isosceles triangles of the Zeiss-Dwyidag model and also connecting the two subsequent rings. The support members are connected to the ring members by a screwing action of each member into spherical ball joints. This allows for easy assembly/disassembly of the structure. The material for the joints and members is GY-70, a high strength, low density composite.

The materials handling subsystem performs all of its tasks with the use of a modified inverted Stewart platform. The modification is a truss structure versus a solid structure. It is supported at three locations on the fifth ring of the dome by six winches, two winches at each location. This allows movement of large loads in six degrees of freedom. The platform is controlled by a D/A position servo and a zero phase error tracking algorithm (ZPETA) prefilter.

An I-beam attached to the platform serves as the track for loading and unloading of all equipment. A modified Hayward grab bucket is used to grasp and handle the equipment. Square notches where the jaws meet are used to precisely mate the bucket connections. The jaws are operated by a simple two-position control action.

To allow for different operations on the platform I-beam, another stationary I-beam is used as a switchtrack subsystem. The platform interfaces with this switchtrack to obtain the different operating tools such as a motor or grab bucket. Another requirement of the materials handling subsystem is to deliver the mined ore out of the dome. This is done by a chute fastened to

the dome structure that is angled to allow the inverted Stewart platform to unload.

After the inverted Stewart platform is set up, drilling operations occur under the structure to locate the ore deposits. The drilling rig has four components: drilling rig platform, turret, robot arm, and drill string. The drilling rig platform is a triangular truss structure that provides the foundation for the rest of the equipment. The turret is positioned on the platform, and provides rotary motion for the robot arm and angular drilling motion for the drill string. The robot arm consists of three links, which enable it to retrieve the drill string sections, connect them together, and provide rotary motion. The drill string runs through the center of the turret and the robot arm provides the rotary motion for drilling operations. The drill string is made of 5 meter sections that are coupled together with treads. Drilling operations can be performed in a vertical orientation up to $\pm 15^\circ$. Cuttings from the drill string are carried to the surface via a stepped auger within the drill string casing and are collected in a material removal container attached to the top of the drill string. The inverted Stewart platform unloads the cuttings.

Once the ore is discovered under the surface, the drilling equipment is moved and the mining equipment is established. The mining equipment consists of explosives, a motor, gripper, modified auger fitted with a tube that prevents material spillage, and bearing cables. The ore is fragmented into small particles with explosives for easy transport to the surface. After the explosives, the gripper is placed around the mining hole to serve as a locking mechanism to attach/detach the auger sections. The auger is made of seven 15 meter sections. The first section has an added drill bit and every subsequent section is connected by threads. The motor is connected onto the inverted Stewart platform and provides the rotary motion for the auger. The motor also connects each auger section as the mining hole increases in depth. Bearing cables are attached to the motor and the ground to apply the downward force needed during mining. The material is brought to the surface by using the downward force of the auger to propel the material to the surface. The mining operation is capable of producing $11.25 \text{ m}^3/\text{hr}$.

To perform all of these actions, a plan was devised to construct the structure. The structure is constructed by making one ring at a time with the Enabler, starting with the top ring, and then using a 47.5 meter vertical May Pole to lift the structure to add each additional ring. This process is continued until the bottom ring is constructed and the structure is finalized. The Enabler interfaces with the pieces by using a three-fingered Stanford/JPL Hand mounted on a Powered Wrist Joint. Cameras mounted on the Enabler provide the feedback during construction. With this process, the structure is assembled in approximately 20 hours.

With this report, a lunar vertical-shaft mining system is described. This preliminary design will greatly advance the possibilities for research of both space and lunar environments. Each subsystem is briefly described, and reference to the reports should be made for further understanding.

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1.1 Overview

A structure is needed to provide support for exploration and mining equipment in the lunar environment. The structure should be a geodesic-dome-shaped design of either hemispherical, parabolic or some similar geometry. The structure is to fit within crater sizes ranging from 30 to 100 meters and must be capable of supporting and interfacing with the materials handling equipment. The beam and joint technology used for the structure should utilize existing space materials technology similar to that used for the Space Station Freedom. The dome must be easily assembled and disassembled so that it can be moved from one location to another. Assembly will be accomplished using a may pole device that will raise the dome and build its subsequent sections from the top down. The structure is to be light weight, cost effective and easy to assemble by a robot.

1.2 Problem Statement

1.2.1 Structural Specifications

- the dome diameter should be approximately 30 meters at the base to accommodate crater sizes ranging from 30 to 100 meters in diameter
- structure should support and successfully interface with Stewart platform
- the interfacing locations of the Stewart platform must be at an optimal height taking into account the length of the material being moved by the platform
- location of the loading points should be chosen to optimize the area accessed by the Stewart platform
- design an aperture in the dome through which the Stewart platform can transport material
- the structure geometry should be hemispherical or parabolic in nature
- structure assembly should be accomplished with the aid of the may pole
- the structure is to be light weight and cost effective

1.2.2 Mobility Specifications

- transport of the structure to space should be accomplished on a single shuttle flight
 - payload weight limitation 949,000 kg
 - cargo bay dimensions 18 m in length and 4.6 m in diameter
- structure is not intended to be stationary; assembly and disassembly should be easily accomplished
- structure is intended to be re locatable

1.2.3 Technical Specifications

- utilize Space Station Freedom technology for beams and joints
- the design must be compatible with the lunar environment in which it will function, specifically the lunar dust and the extreme temperature changes. (temperatures can range from 130°C to -173°C.)
- under maximum loading the maximum bending found in any one beam should be approximately one-half of a foot (15.4 cm)

1.3 Dome Design Selection

1.3.1 Synopsis

Our selection is for a dome based on the parabola $x^2/5$ with a height of 45 meters and radius of 30 meters at the base. The dome is made of nine evenly spaced rings 5 meters vertically apart with a center hole comprising the tenth ring of 1 meter radius (.06 meters from the top). Each ring will be constructed of 14 equal length members with 2 identical support members each forming the isosceles triangles of the Zeiss-Dwyidag classical model.

Loading of the dome takes place at 3 equidistant points on the fifth ring. To the ends of each member are attached sleeves with tapered ends which screw into spherical ball joints to connect to one another. The member beams would be of a hollow equilateral triangular cross section, with an outer side measuring .045 meters and inner side measuring .041 meters, tapering to the inscribed circular cross section for use with the sleeves. Both the beams and joints are to be made out of the composite GY-70, chosen for its outstanding strength, small thermal expansion, and small density.

The important properties of the dome are as following:

- axial shortening = $8.17\text{E-}5$ m
- bending distance = .123 m
- compression ratio, $R = 6.01\text{E-}3$
- critical load (for buckling) = 9801.48 N
- maximum recommended load = 8460 kg
- thermal expansion (for longest member) = -.0002 m
- weight of support members, sleeves, and ball joints = 1988 kg

1.3.2 Type of Geodesic Design

A parabolic dome design was chosen to provide support for the lunar mining expedition. Three possible dome design options exist: elliptical, spherical and parabolic. The elliptical design was found to be inferior to the other two designs due to its inherent buckling tendencies. Of the remaining two designs, parabolic structures were found superior to spherical designs, "...due to their sharper curvature at the top," which distributes more of the load into the vertical plane where the dome is strongest. [3]

1.3.3 Governing Equation

The governing equation selected for the parabolic design is

$$y = 45 - \frac{x^2}{5}.$$

chosen to give our desired dimensions and interior coverage. Smaller constants in the denominator of the equation result in dome geometries of excessive heights, while larger constants flatten the dome, defeating the purpose of utilizing a parabolic design. In Figure 1 of Appendix A is a graphical representation of the dome's parabolic shape resulting from the above equation.

1.3.4 Truss Support

The Zeiss-Dwyidag classical design utilizing isosceles triangles was chosen for the dome's infrastructure. Three design options existed for the dome's infrastructure: the Schwedler and Zeiss-Dwyidag, both considered classical designs (see Figure 1 below), as well as the tetrahedral uni-body design. Due to a larger number of differently orientated supports, the Schwedler and Zeiss-Dwyidag designs have been found to be superior to the tetrahedral uni-body design, thus narrowing our design options down to two. The Zeiss-Dwyidag design was chosen for the following two reasons:

- support members are of equal length (which provides simplicity in manufacturing and construction)
- support members carry equal loads (resulting in uniform fatigue and a simplified strength analysis)

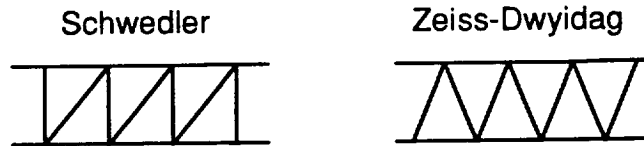


Figure 1

1.3.5 Dome Dimensions

Based on our needs and the given specifications, the base diameter of the dome of 30 meters was chosen. This dimension, when coupled with the governing equation chosen for the parabolic design, results in a overall dome height of 45 meters. Other critical factors considered in choosing these dimensions were:

- effective coverage area of the Stewart platform
- keeping height and radius of the dome close to one another

1.3.6 Components of the Dome

The dome consists of 9 rings equally spaced vertically 5 meters apart (to simplify interfacing of other operations), with 14 support members making up each ring. The influencing factors in these decisions were:

- number of members
- assembly time required
- member length (for bending and buckling)
- mass of the structure
- strength of the structure

To facilitate construction using a may pole, the dome also has a 1 meter radius center hole at the top of the structure, which serves as the tenth ring. Necessitated by the small size of the ring, the center hole will be fabricated in one piece and transported as a whole to the lunar surface.

Each of the 14 beams within a ring has two supports which connect to the ring above it. Therefore the connection of any two rings employs 28 truss supports.

Table 1

ring number	1	2	3	4	5	6	7	8	9	10
base member length (m)	6.732	6.347	5.937	5.497	5.018	4.488	3.887	3.173	2.244	0.449
recessed angle (deg)	9.736	10.352	11.104	12.047	13.283	15.000	17.632	22.500	38.071	
support member length (m)	5.984	5.886	5.789	5.695	5.606	5.529	5.481	5.527	6.355	
support member angle (deg)	57.972	59.713	61.659	63.862	66.405	69.422	73.173	78.287	87.976	

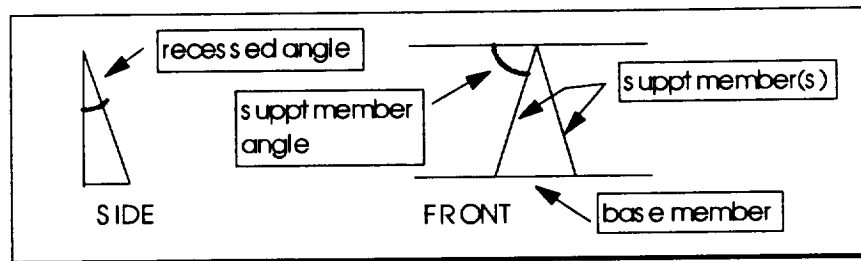


Figure 2

1.4 Member Selection

An equilaterally triangular cross section was chosen for the dome's support members. The critical buckling load for beams of this cross sectional geometry is higher than for any other cross sectional geometries of the same area. Hence, the equilaterally triangular cross section provides the greatest strength to weight ratio of any cross sectional geometry. [1,2]

The actual beam members are hollow, constructed with an outer side length of 4.5 cm with an interior side length of 4.1 cm, chosen on the basis of maximum bending distance of half a foot (15.4 cm) under maximum loading using the equations determined in Appendix C. To interface with the sleeves which connect support beams to the ball joints, the final 18 cm at the end of each beam will taper down into a circular cross-section with a radius of 1.3 cm (determined as the largest circle inscribed by the outer equilateral triangle). The length represents the depth of the sleeve cavity into which each beam is placed with an additional 3 cm clearance. In the final .49 cm of this circular crosssectional length the radius increases to 1.95 cm to provide the lip which supports loading under tension.

1.5 Material Selection

The material to be used in fabricating the structural members, joints, and sleeves for the dome is the composite GY-70. Originally eleven materials were considered, five metals and six composites. These eleven materials were evaluated based on the following criterion: modulus of elasticity (E), coefficient of thermal expansion (CTE), density (ρ) and ultimate tensile strength (F_{tu}). The decision matrix for this evaluation is shown in Tables 1, 2, and 3 of Appendix F. The use of this matrix narrowed the field of materials down to six (three metals and three composites). The performance of remaining six materials were evaluated based on the following measures:

- compression ratio - the ratio of the actual load carried by a structural member of the dome to the maximum load that can be carried by that member
- mass of the dome
- bending in dome that results from loading due to Stewart platform
- thermal expansion of the structural members
- critical load corresponding to buckling

Equations for determining these values were developed in Appendices B through F and used for our spreadsheet calculations found in Table 2 of Appendix F. From this analysis the composite GY-70 was found to be the optimum material choice.

1.6 Joint Design

The joints used in connecting individual support members together are similar to those that are currently used on Space Station Freedom. Each joint is spherical in design and is custom drilled to accommodate for the threaded ends of the supports that attach to it. The positions of the threaded holes in the joints are based on entry angles determined in the dome design (found in Table 1).

The threaded portion of the support members that screws into the joint is a two piece sleeve constructed of symmetrical halves. The halves when fitted together encompass the end of the beam and are free to rotate. The halves are held together by an adhesive. Both the sleeve and ball joint is fabricated out of the composite GY-70 to provide uniform thermal expansion with respect to the connecting members.

The joint is a hollow sphere with an outer radius of 6.4 cm and an inner radius of 3.7 cm. The outer diameter was chosen to provide a minimum arc length of at least 6 millimeters between support connections in the first nine rings. The determination of this distance is based on the angles between structural members (calculations found in Appendix F). Threaded holes are drilled into the spheres for the tapered ends of each connecting beam (4 holes for the 1st ring joints, 6 holes for all other rings). The threads on the sleeves are to industry. The body of the sleeve is cylindrical with an outer diameter of 5 cm with a cylindrical cavity with a diameter of 2.8 cm, and will be attached to the support members prior to spaceflight. The length of the sleeve before the beam cavity measures 10 cm in length, while the length in conjunction with the cavity measures another 15 cm. Drawings of both the joint and sleeve can be found in Figures 3 and 4 of Appendix A.

Joints on the pre-fabricated tenth ring consist of cylindrical segments with a length of 6.4 cm and a radius of 4 cm. Into this segment is drilled the corresponding hole to industry standards for the insertion of one support member. These cylindrical segments, one for each support member to be connected, will be arranged about the tenth ring at the appropriate angles of alignment (see Table 1). The reason for the use of cylindrical segments rather than spherical joints, is that the radius of a spherical joint needed to provide the necessary arc length clearance would be more than six times larger than those designed above. Larger ball joints would detrimentally increase the weight of the dome.

1.7 Interfacing With Other Operations

1.7.1 Materials Handling

In order to remove mined materials from within the dome, an area (2 meters wide by 3.25 meters high) between support members must be accessible. This represents the size of the clamps/jaws which collect regolith and other materials brought up from beneath the moon's surface. The area found between the supports of the dome's third ring are 2 meters wide by 3.67 meters high, meeting these requirements.

An additional concern in material handling is the relative amount of coverage available to the Stewart Platform. To perform efficiently, the Stewart Platform needs to be able to cover at least 20 percent of the surface area enclosed within the dome. By tailoring the dome's parabolic equation to meet this need, a coverage of nearly 23% is available.

1.7.2 Mining

To help provide downward thrust for the Stewart Platform's mining operations, two motors will be attached to the dome at distant points on the third ring (10 meter vertical height) and connected to the platform with cables. As a result, two new loading points are created on the dome, with a resulting total load based on these motors of 750 Newtons. Use of the equations derived in Appendix B show that these motors convey a maximum resultant force of only 80 Newtons to adjacent supports below the third ring. Due to the dividing nature of the load inherent in the use of a geodesic dome, the maximum resultant force upon the 2nd ring supports is 280 Newtons, 1/3 the maximum load conveyed by the Stewart Platform itself on support members below the 5th ring. Therefore, this additional force presents no threat to the dome's integrity.

1.7.3 Assembly

The assembly of the dome structure is to be accomplished with the aid of a May Pole and the Enabler robot. To attach the structure to the May Pole itself, a stable, pre-constructed interface with the dome and an aperture of a one meter radius were needed. This was accomplished through the use of a uni-body center hole with the required radius.

The dome design must take into consideration the capabilities of the robot Enabler. Placing the dome rings at 5 meter intervals places all connections within its 40 foot reach. The length of the sleeves was also determined in conjunction with the 8-10 inch width of the Enabler's grip.

1.8 Future Work

1.8.1 Adhesive for Sleeve Assembly

The sleeve which fits about each end of a support member of the dome is a two piece assembly. The "male" protrusions from the interior surface of each piece in conjunction with the parallel "female" receptacles of each serve to hold the sleeve together. In addition, an adhesive should be applied to the inner surfaces of each half of the sleeve. This adhesive must be strong as well as resistant to the temperatures encountered on the lunar surface.

1.8.2 Center Hole Design

The angles at which support members approach the tenth ring is approximately 4 degrees, which poses a problem of clearance at the ball joints. Considering this difficulty and small size of the center hole (one meter radius), it was decided the ring should be pre-fabricated and shipped to the mining site in an already

assembled fashion. A uni-body ring with segmented joints located at the proper angles appears to be the most efficient solution to both problems. Although no detailed analysis has been completed, using a cylindrical cross section with area similar to that found in the equilateral triangular cross section used in the support members should prove to be an adequate design for these joints.

1.8.3 Mounting Brackets

Positioning of the mounting fixtures for the Stewart platform control motors has been specified. The detailed design of these fixtures, though, has not been completed. It is recommended that the fixtures be designed as plates that are attached to the dome at the specified load points. The plates would be held in place using brackets that attach each plate to the structural members extending from the joint corresponding to the plate's position. The motors would then be attached to the plates.

1.8.4 Redesign for Weight Minimization

When the infrastructure of the dome (number of rings, number of supports) was optimized to minimize mass, it did not include a consideration of the impact that the mass of the sleeves and joints would play. Having performed such analysis, it is now possible that a new arrangement of the dome may be more appropriate (i.e. a smaller weight). It is predicted that using a smaller number of rings and supports, thereby reducing the number of sleeves and joints, will result in a dome with a significantly smaller mass (i.e. 6 rings with 10 supports per ring).

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Appendix A

Drawings

List of Figures

- A1 Dome
- A2 Support Member
- A3 Ball Joint
- A4 Sleeve

APPENDIX A DRAWINGS

DIMENSIONS

HIGHT: 45 meters

BASE DIAMETER: 30 meters

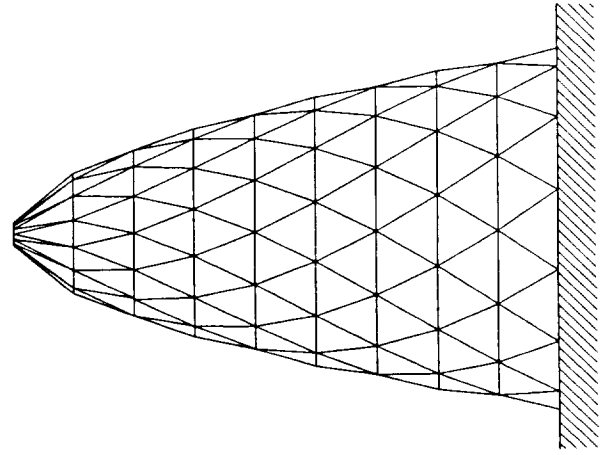
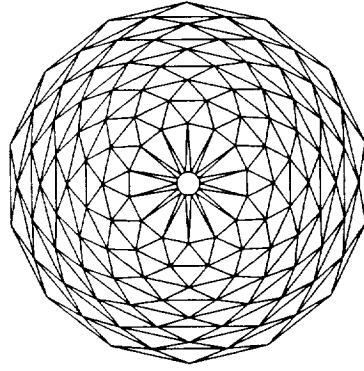
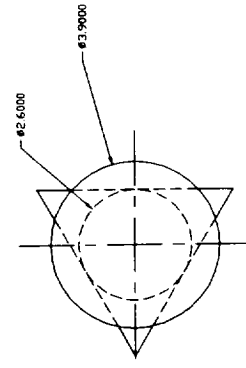
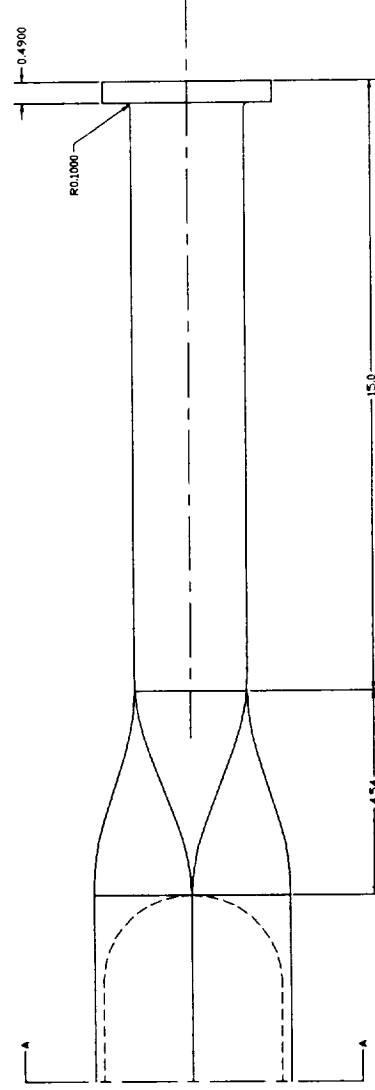
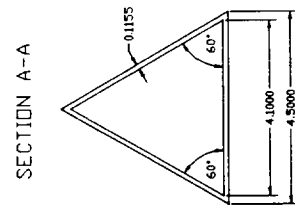


FIGURE 1

Project Name	Project Number	Project Date
Project Location	Project Status	Project Budget
Project Manager	Project Engineer	Project Designer
Project Architect	Project Contractor	Project Supplier
Project Client	Project Consultant	Project Inspector
Project Owner	Project Operator	Project Maintainer

FIGURE 2
SCALE 1:1

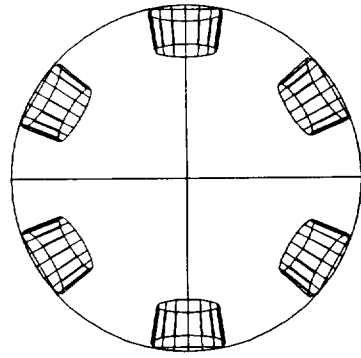
MATERIAL: GRAPHITE COMPOSITE GY-70



Project Name	Structure Group	Scale	1:1
Project Number	Structure Group	Scale	1:1
Project Name	Structure Group	Scale	1:1
Project Number	Structure Group	Scale	1:1

FIGURE 3

SCALE 1:1



MATERIAL: GRAPHITE COMPOSITE GY-70

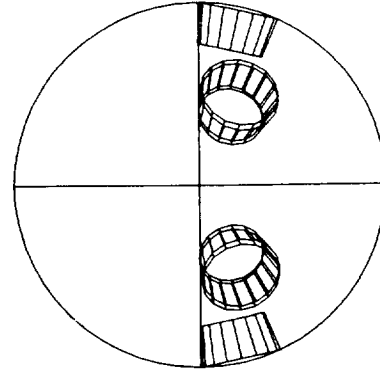
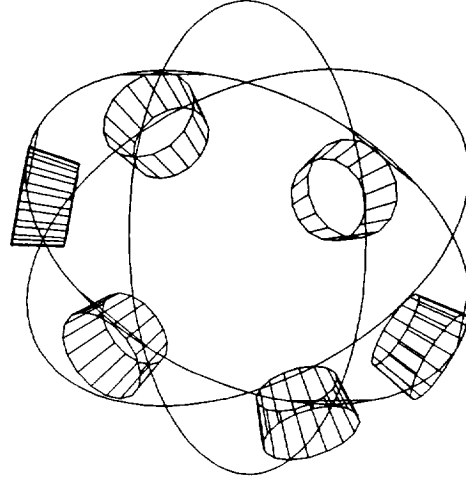
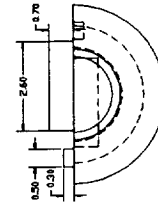
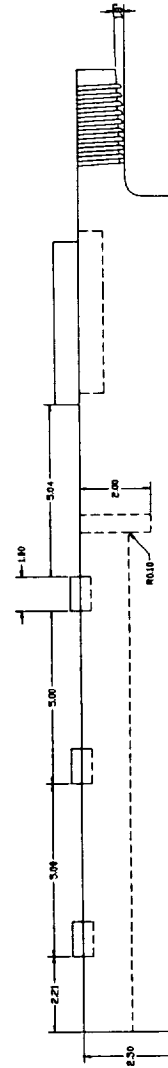
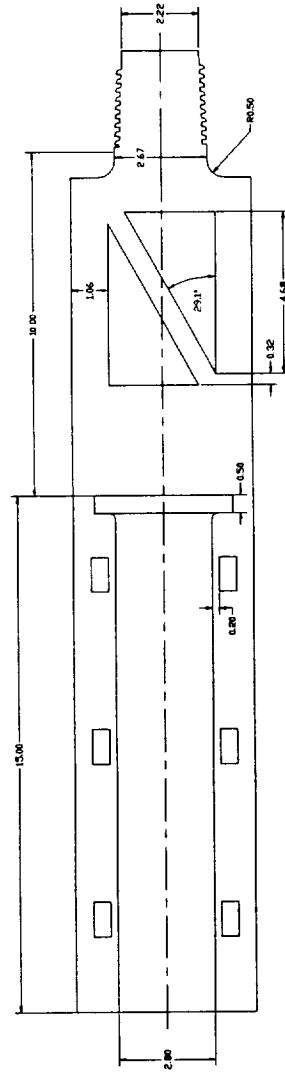


FIGURE 4

SCALE: 1:1

MATERIAL: GRAPHITE COMPOSITE GY-70



Appendix B Compression Analysis

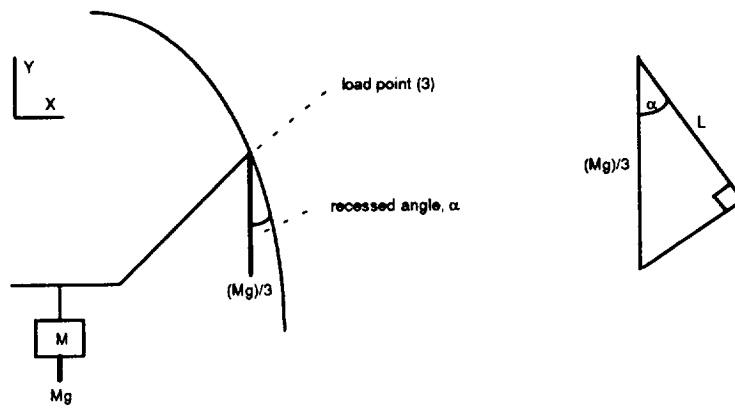


Figure B-1

$$L = \frac{mg}{3} \cos \alpha$$

ASSUMPTION: Due to the fact that the cables are loaded at equal angles at both the dome and the platform, component of the load in the Y-direction is equal to the total load divided by the number of equidistant load points.

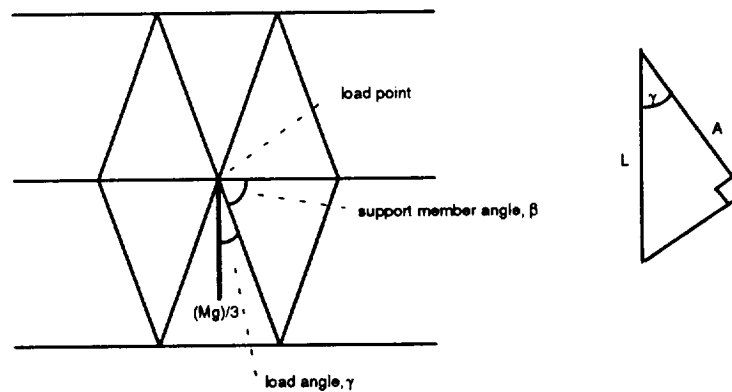


Figure B-2

$$A = L \cos \gamma = \frac{mg}{3} \cos \alpha \cos \gamma$$

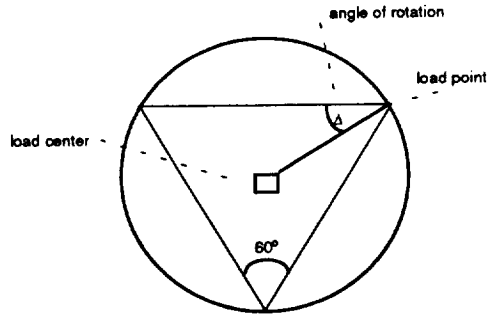


Figure B-3

ASSUMPTION: Since $\Delta > \alpha$ at load points, compression in ring members must be less than that in support members.

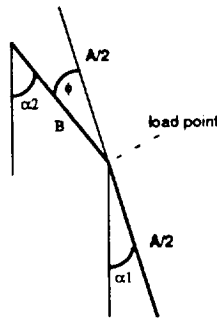


Figure B-4

If support members were on the same plane ($\alpha_1 = \alpha_2$), the load would be equally distributed (tension above, compression below). But $\alpha_2 > \alpha_1$ therefore,

$$B = \frac{A}{2} \cos \phi = \frac{A}{2} \cos(\alpha_2 - \alpha_1)$$

ASSUMPTION: Since $B < A$, the tension in the support members above the load point is less than the compression found in the lower support beams.

ASSUMPTION: Since support member geometry is symmetrical, adjacent members will divide the load. Once removed from the supports directly in contact with the load points the load on each member is decreased by $1/2$, and so forth throughout the structure. Therefore, the greatest compressive load will be found in those support members in contact below the load points.

$$load = \frac{A}{4} + \frac{A}{4}(1 - \cos \phi) = \frac{mg}{12} \cos \alpha \cos \gamma (2 - \cos \phi)$$

For our maximum recommended load, the greatest compressive force seen by any single support member was calculated to be 1028.51 N.

Calculations of the axial shortening due to this load are found using the following equation:

$$\delta = \frac{FL}{EA}$$

F = The loading Force in Newtons

L = The length of the beam

E = The modulus of elasticity of the material

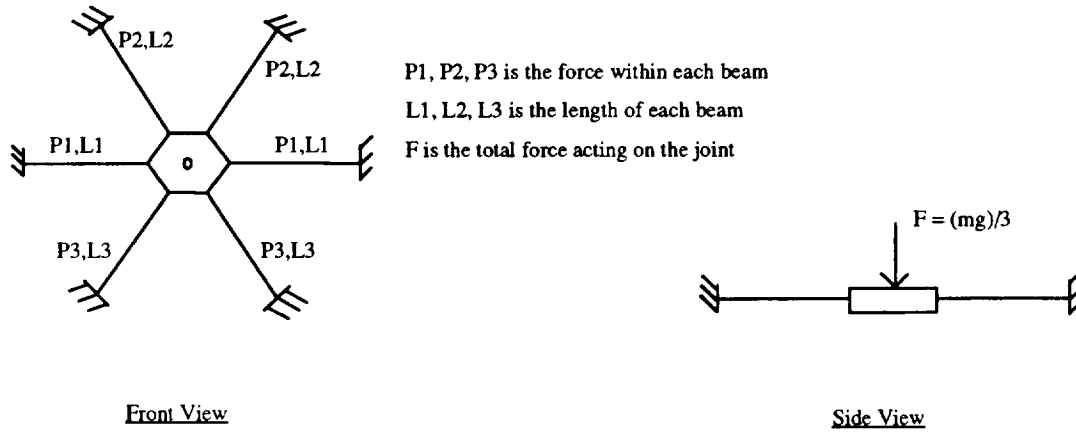
A = The cross - sectional area of the beam

The greatest axial shortening found within the dome is 8.17E-5 m. Since the beam compression was this small it was ignored.

Appendix C

Bending Analysis

The Stewart Platform interfaces with the dome structure in three locations. At each of these locations the connection occurs at a joint. The greatest amount of bending occurs at the joint interfacing with the Stewart platform. The following calculation is the worst case scenario for bending at the Stewart Platform joint interfacing:



Assumptions: The six beam members are in the same plane. The load acting on the joints acts totally in the direction perpendicular to the joint and results in pure bending of the six beams. Assumed none of the forces are translated into axial loading of the beams in order to obtain a conservative estimate with a factor of safety.

F is the total force in Newtons acting of the joint. It is the summation of the weight of the Stewart platform itself and the maximum load the platform will ever carry.

Assumptions: The deflection of the beams was assumed to be equal since the joint deflects by a certain amount. The following is the equation for equal deflection for each beam.

$$\delta = \frac{P_1 L_1}{3EI} = \frac{P_2 L_2}{3EI} = \frac{P_3 L_3}{3EI}$$

$$P_1 L_1 = P_2 L_2 = P_3 L_3$$

Solving for P_2 and P_3 in terms of P_1 , produces the following equation:

$$P_2 = \frac{L_1}{L_2} P_1 \quad \text{and} \quad P_3 = \frac{L_1}{L_3} P_1$$

The following equation is the sum of the forces acting on the six beams.

$$P_1 + P_1 + P_2 + P_2 + P_3 + P_3 = F = \frac{mg}{3} \rightarrow 2P_1 + 2P_2 + 2P_3 = F = \frac{mg}{3}$$

Substituting for P_2 and P_3 in terms of P_1 into the previous equation produces:

$$\frac{L_1}{L_2} P_1 + \frac{L_1}{L_3} P_1 + P_1 = F = \frac{mg}{6}$$

Solving for P_1 results in the following equation:

$$P_1 = \frac{mg}{6} \left[1 + \frac{L_1}{L_2} + \frac{L_1}{L_3} \right]^{-1}$$

Likewise the equations for P_2 and P_3 were solved in a similar manor:

$$P_2 = \frac{mg}{6} \left[1 + \frac{L_2}{L_1} + \frac{L_2}{L_3} \right]^{-1}$$
$$P_3 = \frac{mg}{6} \left[1 + \frac{L_3}{L_1} + \frac{L_3}{L_2} \right]^{-1}$$

The forces acting each of the beams is different, however, by solving the following equation using the formula for P_1 found above the maximum deflection of the joint can be determined.

$$\delta = \frac{P_1 L_1}{3EI}$$

A design specification required that the maximum bending deflection be 0.154 m. Using a factor of safety of 1.25 in the maximum loading, the greatest deflection seen by any individual support member is 0.123 m. The support member dimensions were chosen such that this requirement was met.

Appendix D

Buckling Analysis

The critical load at which buckling occurs can be found from Euler's equation for buckling.

$$P_{critical} = \frac{\pi^2 EI}{L^2}$$

E - modulus of elasticity L - length of column
I - moment of inertia

The preceding equation applies to columns that are pinned at both ends. For columns with different end conditions, an effective length, L_e , is employed. For a column that is fixed at both ends,

$$L_e = \frac{L}{2}$$

Euler's equation for buckling becomes,

$$P_{critical} = \frac{\pi^2 EI}{L_e^2}$$

Substituting for L_e yields,

$$P_{critical} = \frac{4\pi^2 EI}{L^2}$$

The critical load associated with the GY-70 composite and the support beam design is 9801.48 N, over nine times the maximum load any single support member undergoes. Therefore, buckling is not a consideration.

Appendix E

Thermal Analysis

The following equation calculates the expansion or contraction of a material due to temperature changes:

$$\delta_T = CTE (\Delta T) L$$

$$CTE = \text{The coefficient of thermal expansion} \left[\frac{1}{^\circ\text{F}} \right]$$

$$\Delta T = \text{The change in temperature} [^\circ\text{F}]$$

$$L = \text{length of the material at } T_0 \text{ [m]}$$

The temperature range of the moon is from 266° F to -280° F. The beams are made of a graphite fiber with a coefficient of thermal expansion of -0.626E-6. The longest beam length is 6.732 m. Using 60° as the base temperature the greatest expansion due to changes in temperature is 1.43 mm and the greatest compression by this beam would be 0.868 mm. The total change in length of the longest beam within the dome is therefore 2.3 mm and is negligible when considering the tolerances between beams.

Appendix F

Material Selection

Construction of Decision Matrix

Eleven materials in all were considered for use in the design of the domes structural members. The materials considered were rated according to the following criterion:

- E - modulus of elasticity
- CTE - coefficient of thermal expansion
- p - density
- Ftu - ultimate strength

A decision matrix was constructed to compare the merits of each of the possible materials. Using aluminum as the datum, the following relations were used to rate the various materials (the subscript Al represents aluminum):

- For modulus of elasticity:
$$E_{rated} = \frac{E_{material}}{E_{Al}}$$
- For CTE:
$$CTE_{rated} = \frac{CTE_{Al} - CTE_{material}}{CTE_{Al}}$$
- For density (p):
$$p_{rated} = \frac{p_{Al} - p_{material}}{p_{Al}}$$
- For Ftu:
$$Ftu_{rated} = \frac{Ftu_{material}}{Ftu_{Al}}$$

The weight assigned to each of these rated values could be changed within the decision matrices as seen on the following pages. Three principal combinations of weighted criterion were evaluated and each is shown.

Table F-1

Criterion	E	CTE	density (p)	Ftu					
Weight	0.5	0.2	0.2	0.1					
	E-rated	E-wtd	CTE-rated	CTE-wtd	p-rated	p-wtd	Ftu-rated	Ftu-wtd	Wtd Total
Material:									
1) Metals									
Aluminum*	1.00	0.50	0.00	0.00	0.00	0.00	1.00	0.10	0.60
Titanium	1.52	0.76	0.55	0.11	-0.60	-0.12	2.28	0.23	0.98
Steel	2.76	1.38	0.51	0.10	-1.83	-0.37	3.71	0.37	1.49
INVAR	1.95	0.98	0.94	0.19	-1.91	-0.38	0.92	0.09	0.87
Beryllium	4.05	2.03	0.50	0.10	0.34	0.07	0.62	0.06	2.26
2) Composites									
AS	2.66	1.33	1.03	0.21	0.35	0.07	5.57	0.56	2.16
T300	3.04	1.52	1.03	0.21	0.37	0.07	5.00	0.50	2.30
Modmor II	3.80	1.90	1.03	0.21	0.37	0.07	5.71	0.57	2.75
HM-S	4.95	2.48	1.04	0.21	0.32	0.06	4.85	0.49	3.23
Pitch	4.76	2.38	1.04	0.21	0.28	0.06	2.50	0.25	2.89
GY-70	6.76	3.38	1.04	0.21	0.29	0.06	3.57	0.36	4.00

Table F-2

Criterion	E	CTE	density (p)	Ftu					
Weight	0.2	0.5	0.2	0.1					
	E-rated	E-wtd	CTE-rated	CTE-wtd	p-rated	p-wtd	Ftu-rated	Ftu-wtd	Wtd Total
Material:									
1) Metals									
Aluminum*	1.00	0.20	0.00	0.00	0.00	0.00	1.00	0.10	0.30
Titanium	1.52	0.30	0.55	0.28	-0.60	-0.12	2.28	0.23	0.69
Steel	2.76	0.55	0.51	0.26	-1.83	-0.37	3.71	0.37	0.81
INVAR	1.95	0.39	0.94	0.47	-1.91	-0.38	0.92	0.09	0.57
Beryllium	4.05	0.81	0.50	0.25	0.34	0.07	0.62	0.06	1.19
2) Composites									
AS	2.66	0.53	1.03	0.52	0.35	0.07	5.57	0.56	1.67
T300	3.04	0.61	1.03	0.52	0.37	0.07	5.00	0.50	1.70
Modmor II	3.80	0.76	1.03	0.52	0.37	0.07	5.71	0.57	1.92
HM-S	4.95	0.99	1.04	0.52	0.32	0.06	4.85	0.49	2.06
Pitch	4.76	0.95	1.04	0.52	0.28	0.06	2.50	0.25	1.78
GY-70	6.76	1.35	1.04	0.52	0.29	0.06	3.57	0.36	2.29

Table F-3

Criterion	E	CTE	density (p)	Ftu					
Weight	0.2	0.2	0.5	0.1					
	E-rated	E-wtd	CTE-rated	CTE-wtd	p-rated	p-wtd	Ftu-rated	Ftu-wtd	Wtd Total
Material:									
1) Metals									
Aluminum*	1.00	0.20	0.00	0.00	0.00	0.00	1.00	0.10	0.30
Titanium	1.52	0.30	0.55	0.11	-0.60	-0.30	2.28	0.23	0.34
Steel	2.76	0.55	0.51	0.10	-1.83	-0.92	3.71	0.37	0.11
INVAR	1.95	0.39	0.94	0.19	-1.91	-0.96	0.92	0.09	-0.29
Beryllium	4.05	0.81	0.50	0.10	0.34	0.17	0.62	0.06	1.14
2) Composites									
AS	2.66	0.53	1.03	0.21	0.35	0.18	5.57	0.56	1.47
T300	3.04	0.61	1.03	0.21	0.37	0.19	5.00	0.50	1.50
Modmor II	3.80	0.76	1.03	0.21	0.37	0.19	5.71	0.57	1.72
HM-S	4.95	0.99	1.04	0.21	0.32	0.16	4.85	0.49	1.84
Pitch	4.76	0.95	1.04	0.21	0.28	0.14	2.50	0.25	1.55
GY-70	6.76	1.35	1.04	0.21	0.29	0.15	3.57	0.36	2.06

* Datum

Appendix G

Joint/Sleeve Analysis

Ball Joint Inner and Outer Radius Determination

The following equation calculates the critical radius for the ball joint based on the cord length and the angle between two members:

$$r = \frac{L}{2} \left(\frac{1}{\sin \frac{\alpha}{2}} \right)$$

where r is the radius of the ball, α is the angle between two supports and L is the cord length.

$$r = \frac{5.1}{2} \left(\frac{1}{\sin \left(\frac{23.44}{2} \right)} \right) = 12.55$$

The minimal distance for the outside diameter of the ball joint is 12.55 cm. Because of the taper of the screw the actual area occupied by the connection is small than the cone used in the analysis.

From industry standards the depth of the drilled hole needs to be 3.11 cm, therefore we chose the inside diameter of the sphere to be 9.4 cm.

Member lip diameter analysis

In a tensile loading the lip of the support member carries the load. The following formula calculates the critical area of the lip necessary to prevent shearing.

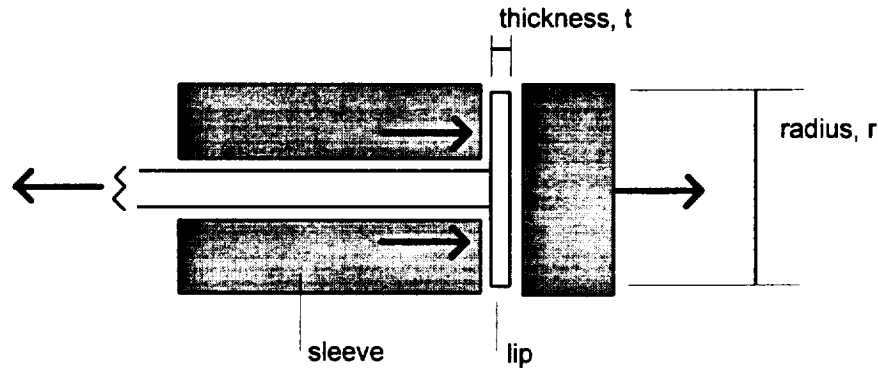


Figure G-1

$$A = \frac{P}{F_u} = \frac{806.8 N}{1.723 \times 10^9 \frac{N}{m^2}} = 4.684 \times 10^{-7} m^2$$

Where P is the maximum tensile load encountered in the dome.

$$t = \frac{A}{2\pi r}$$

$$t = \frac{4.6839 \times 10^{-7} m^2}{2\pi \times 0.045 m} = 1.66 \times 10^{-6} m$$

Where r is the radius of the circular beam cross-section and t is the thickness of the lip. For convenience the thickness of the lip to be 2 cm and 1.95 cm radius for the lip.

Appendix H Spread Sheet

The following tables represent the spreadsheet modeling used to easily calculate all desired measurements and values necessary to the dome design and material selection.

Table 1 represents the calculations of member lengths and angles. **It is important to understand that the member lengths listed are without compensating for the sleeves.** Therefore, in order to determine the ACTUAL lengths of the support beams, subtract .3 m to compensate for the sleeve lengths at each end. NOTE: This is already done for all calculations found in Table 2.

Table 2 represents the calculations of the strength characteristics of the individual support members as well as the second material selection decision matrix. In calculating these values, certain conversion factors were used. They are:

E & Ftu	lbf to Newtons	x 6890
density	lb/in ³ to kg/m ³	x 27637.267

It also should be noted that the values of the coefficients of thermal expansion are x 10⁻⁶.

Finally, the calculation of the moment of inertia for a hollow equilateral cross-section is found to be:

$$I = \frac{\sqrt{3}}{96} (l_o^4 - l_i^4)$$

Table H-1
Dome Calculations

governing equation	expt=	2
	num=	5

# of rings	9
vert. ring height (m)	5
# of members in base	14

ring #	1	2	3	4	5	6	7	8	9
base radius (m)	15.000	14.142	13.229	12.247	11.180	10.000	8.660	7.071	5.000
base circumference (m)	94.248	88.858	83.119	76.953	70.248	62.832	54.414	44.429	31.416
base member length (m)	6.732	6.347	5.937	5.497	5.018	4.488	3.887	3.173	2.244
abs. ring distance (m)	5.073	5.083	5.095	5.113	5.137	5.176	5.246	5.412	
recessed angle	9.736	10.352	11.104	12.047	13.283	15.000	17.632	22.500	
suppl member length (m)	5.984	5.886	5.789	5.695	5.606	5.529	5.481	5.527	
suppl member angle	57.972	59.713	61.659	63.862	66.405	69.422	73.173	78.287	

center hole	1
base radius (m)	0.067
dist down from peak (m)	6.283
base circumference (m)	0.449
base member length (m)	
top ring	
abs. ring distance (m)	6.351
recessed angle	38.071
suppl member length (m)	6.355
suppl member angle	87.976

Table H-2
Material Calculations

compression analysis	
triangular beam	I (out) 0.045 I (in) 0.041
cross sectional area	0.00015
load mass (kg)	8460
number of load pts	3
loaded ring	5
recess angle	13.283
ress angl (ring + 1)	15.000
suppt mbr angle	66.405
avg mbr lngth (m)	4.910
max comp force (N)	1028.510
bending analysis	
length.1 (m)	5.018
length.2 (m)	5.606
length.3 (m)	5.695
P.1 (N)	830.423
P.2 (N)	743.263
P.3 (N)	731.664
moment of inertia, I	2.3E-08
thermal analysis	
high temp (deg far)	266
low temp (deg far)	-280
lngst beam lgth (m)	6.732

material	Aluminum	Steel	Titanium	Modmor II	HM-S	GY-70
name						
E (ms)	10.5	29	16	40	52	71
Ftu (ksi)	70	260	160	400	340	250
density	0.1	0.283	0.16	0.063	0.068	0.071
CTE	13	6.3	5.8	-0.5	-0.61	-0.626
weight						
comp ratio, R	2.15E-02	5.78E-03	9.40E-03	3.76E-03	4.42E-03	6.01E-03
axial shortening (m)	5.53E-04	2.00E-04	3.63E-04	1.45E-04	1.12E-04	8.17E-05
mass of beams(kg)	764	2162	1222	481	520	542
bending dist (m)	0.835	0.302	0.548	0.219	0.169	0.123
axial expansion (m)	0.048	0.023	0.021	-0.002	-0.002	-0.002
critical load (N)	1449.515	4003.421	2208.784	5521.960	7178.548	9801.480
decision total						
	1	0.641	0.677	0.228	0.203	0.181
mass of sleeves	258.187					
mass of joints	1186.929					
comb mass of dome	1988					

Table H-2
Material Calculations

compression analysis	
triangular beam	l (out) 0.045
	l (in) 0.041
cross sectional area	0.00015
load mass (kg)	10580
number of load pts	3
loaded ring	5
recess angle	13.283
rcss angl (ring + 1)	15.000
suppt mbr angle	66.405
avg mbr lngth (m)	4.910
max comp force (N)	1286.246

bending analysis	
length.1 (m)	5.018
length.2 (m)	5.606
length.3 (m)	5.695
p.1 (N)	1038.519
p.2 (N)	929.518
p.3 (N)	915.012
moment of inertia. I	2.3E-08

thermal analysis	
high temp (deg far)	266
low temp (deg far)	-280
lngst beam lgth (m)	6.732

material	Aluminum	Steel	Titanium	Modmor II	HM-S	GY-70
name						
E (ksi)	10.5	29	16	40	52	71
Ftu (ksi)	70	260	160	400	340	250
density	0.1	0.283	0.16	0.063	0.068	0.071
CTE	13	6.3	5.8	-0.5	-0.61	-0.626

comp ratio, R	2.69E-02	7.23E-03	1.17E-02	4.70E-03	5.53E-03	7.52E-03
axial shortening (m)	6.91E-04	2.50E-04	4.53E-04	1.81E-04	1.40E-04	1.02E-04
mass of beams(kg)	764	2162	1222	481	520	542
bending dist (m)	1.044	0.378	0.685	0.274	0.211	0.154
axial expansion (m)	0.048	0.023	0.021	-0.002	-0.002	-0.002
critical load (N)	1449.515	4003.421	2208.784	5521.960	7178.548	9801.480

decision total	1	0.641	0.677	0.228	0.203	0.181
----------------	---	-------	-------	-------	-------	-------

mass of sleeves	258.187
mass of joints	1186.929
comb mass of dome	1988

weight
0.05
0.15
0.1
0.3
0.3
0.1

Section Two:

Materials Handling System

2.0 Introduction

The materials handling system supports lunar crater-based mining by moving mined material outside the structure and by moving mining and drilling equipment inside the structure as necessary. The materials handling system is largely dependent on the needs of other systems in the lunar mining project, and consequently requires interfaces to accommodate them. See drawing MH-W1 in Appendix 2-1 for an overall representation of the system.

2.1 Problem Statement

The materials handling portion of the lunar crater-based mining project was divided into the following subsystems:

- loading and unloading materials from the transporting equipment;
- locating the destination point;
- transporting the materials to the destination point ;
- interfacing with other systems; and
- controlling the actions of all equipment

The main concerns revolved around operation of equipment in a lunar environment. Specifically, the following were considered during design (Gertsch):

- **Vacuum:** In a vacuum, the gaseous layer present on metal parts in an atmosphere does not exist. This can cause parts to weld together over time, affecting many common equipment components. In addition, evaporation of lubricants is accelerated and an additional increase in friction develops, because there is no gas lubrication due to atmosphere. Finally, heat dissipation is by radiation only.
- **Abrasive Electrostatic Dust:** The abrasiveness of lunar dust will cause wear problems, while the electrostatic nature will make avoidance of the dust impossible. Wire ropes and sheaves, chutes, gears, sprockets and chains and contact points will be affected.
- **Temperature Extremes:** The temperature varies from -180°C to 134°C. Materials must handle temperature extremes as well as the rate

of temperature change. Operations may be limited to daylight hours (equivalent to about 14 consecutive earth days at a time) or a heater may need to be in operation nighttime hours. Excessive thermal expansion could cause sagging in cables, etc.

- **1/6 Gravity:** While items are easier to lift on the moon than they would be on earth, they maintain the same inertia.
- **Transportation Costs:** Due to high delivery costs, the weight and volume of the system must be kept at a minimum.

The materials handling design for the project is highly dependent on the designs chosen by the structures team and the mining team. Workable, integrated interfaces have been implemented for the success of this mining operation.

2.2 Material Selection and Recommendation

As with past lunar designs, aluminum 6061-T6 is the material of choice. It combines the light weight of aluminum with the higher strength of a steel. It has improved fatigue properties and higher strength retention at high temperatures as compared to steel. The MMC is lighter than the titanium alloys and has better thermal properties than straight aluminum alloys (Brazell).

2.3 Transport Subsystem

2.3.1. INTRODUCTION

The purpose of the transport subsystem is to provide an effective means of transportation for the location of lunar mining and drilling equipment and for the removal of lunar ore and regolith. The subsystem must be versatile to allow for precise positioning of equipment. It must also be of minimal weight and simplistic in design.

2.3.2. RECOMMENDATION

The inverted Stewart platform will best meet the specifications described above. The inverted Stewart platform is a crane system which is composed of a stable triangular platform supported by six cables. It is capable of precisely positioning

large loads in six degrees of freedom. It will be supported at three locations on the dome by six winches. The winches will be attached to the structure through means specified by the structures team. The recommended material for the platform system is aluminum 6061-T6. The platform will consist of a modification involving the attachment of an I-beam along the base of the platform. The I-beam will serve as a track for the loading and unloading of equipment on and off of the platform. An additional modification involves building the platform as a truss structure as opposed to a solid triangular plate. The mass of the platform will be approximately 163 to 177 kg and measure 6 meters to a side.

The platform will be supported at a distance of 20 meters above the bottom of the structure. The three supports will be equally spaced 120° apart. Two winches will reside at each support and will control one cable each. The cables will attach by ball and socket joints to the corners of the platform. Each corner of the platform will have two cables attached to it by ball and socket joints. The recommended cable diameter is 1.27 cm. A limit of 10° between the platform and its cables has been set.

The platform itself is composed of trusses. The I-beam will be secured by plates which are bolted to the beam and welded to six of the supporting truss members. An outer radius of 2.54 cm and inner radius of 2.22 cm is recommended for all truss members with the exception of the six welded members. These trusses will acquire the properties of Aluminum 6061-T6 which has a yield strength of 55.2 MPa. These members will have a inner radius of 1.27 cm. The beam chosen for the platform is a standard I-beam cross section (Figure 2-4, page 2-11) that is approximately 4 meters in length. The beam will serve as a track for the trolleys of the jaws and the mining equipment.

2.3.3. JUSTIFICATION

Concept: The cable supports along with the truss structure of the platform make this concept the lightest possible feasible solution. In addition, the cables remain in maximum stress a maximum amount of the time giving the system an optimum geometry. The configuration of the cables gives the platform six degrees of freedom. The I-beam allows for a simple method of transferring

equipment and material and uses no power in the process. The recommended platform design is in Appendix 2-1 (see Drawing 2-1). The resulting area coverage for the lower platform is 163 meters². This is approximately 23% of the total area of the base of the dome.

Sample calculations regarding the transport subsystem are in Appendix 2-2.

Material of Construction: The recommended material of construction is aluminum 6061-T6. This material is described in section 2.2.

2.4 Load/Unload Subsystem

2.4.1. INTRODUCTION

The function of the load/unload subsystem is to hold or clamp various materials so they can be transported by the inverted Stewart platform. The main types of materials to be moved are equipment, such as the platform and drill rods used in crater drilling, and ore that is retrieved by the mining operation. These forms of material handling require different functions of the loading/unloading subsystem (i.e., precision handling compared to bulk solids handling); accordingly, the load/unload subsystem must be versatile while maintaining the simplicity and ruggedness necessary for lunar operation.

2.4.2. RECOMMENDATION

A modified Hayward grab bucket will best meet the requirements as described above. The modification consists of adding "square notches" where the clam shell pieces meet, so grab bucket mates can be precisely picked up. The grab bucket will have a mass between 408 kg and 544 kg, with a load rating of 9808 N on the moon. The material of construction is aluminum 6061-T6.

Assuming a load of 9808 N, the jaws require a small drum outer diameter of 23.4 cm with a 20.8 cm inner diameter. With this small drum diameter, a large drum diameter of 47.0 cm provides for minimal power use (2.8 kW). The weight of the entire system (with a non-solid large drum core) will be under 544 kg.

The jaw mates will consist of hollow tubes with outer diameters of 4.6 cm and 4.1 cm inner diameters. With a 10.16 cm shaft, a load of 9808 N can be lifted and tilted to a 40° angle. The shaft will have a head (see Figure 2-1) with a diameter of 9.68 cm. The head will have supporting rods to support the load.

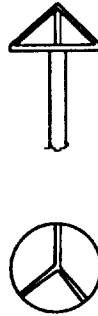


Figure 2-1: The Jaw Mate Head

2.4.3. JUSTIFICATION

The Concept: The Hayward grab bucket is a conventional materials handling design used for picking up and transferring loads of ores, sands, etc. in mining operations. It consists of two clam shells that are rotated about a fixed shaft by a drum assembly.

The modified Hayward grab bucket with the square wave design will allow the grab bucket to pick up mates in an easy fashion with maximum support to the load.

The shafts recommended for the mates simply allow an object to be lifted with no contact occurring between the object and the grab bucket. Thus more delicate operations can be performed than would otherwise be possible with the jaws. The design of the shaft is based on supporting the 9808 N load.

Material of Construction: The recommended material, aluminum 6061-T6, was used on previous lunar missions. It is described in section 2.2. Calculations for verifying the integrity of aluminum 6061-T6 for this application are in Appendix 2-3. The drilling operation's triangular base will be the heaviest object that the materials handling system has to handle. This base will

weigh no more than 1 metric ton on the moon. Thus, the static force analysis on the grab bucket was completed using a 9808 N downward vertical force.

2.5 Locate Subsystem

2.5.1. INTRODUCTION

In order for the materials handling system to function, another subsystem must exist that provides timely feedback concerning the current location of the transport and load/unload subsystems. The locate subsystem not only provides this information, but also the locations of all possible destinations of the aforementioned subsystems.

2.5.2. RECOMMENDATION

SHIRAN (Cubic), a form of short range high accuracy radio navigation, suits all the needs of the locate subsystem. It measures the distances between a transceiver on the inverted Stewart platform and each of four beacons placed on the structure. This operation is accomplished by measuring the phase delay of radio signals transmitted from the beacons. The beacons each transmit four ranging frequencies, 500 kHz, 1.08 MHz, 2.33 MHz, and 5.00 MHz. The locate subsystem's components communicate using 20 MHz coded signals. SHIRAN components are housed in carbon epoxy composite casings, and all necessary antennas are made of aluminum 6061-T6. See Appendix 2-4 for a detailed description of the SHIRAN locate subsystem.

2.5.3. JUSTIFICATION

The SHIRAN system was chosen for its high accuracy of measurement and versatility. Error models have made its measurements nearly perfect (Gyer). In the lunar mining project, it will deliver a resolution of 3 cm. By using the SHIRAN system, the beacons can be placed on the structure instead of in the crater (as would have been required by other radio navigation systems). The power required to operate this system is very small because the transmitters radiate intermittent output. Each signal only consumes about 50 μ W.

The selection of the four ranging frequencies was based on previously published data (Cubic). The frequencies are described by the following titles: very coarse, coarse, fine, and very fine. One must select the very coarse frequency so that the maximum phase delay corresponds to the maximum distance to be measured. In the lunar mining scenario, this will be at most 300 meters. The very coarse frequency selection arises from the data shown in Figure A4-2 in Appendix 2-4. The very fine frequency must be chosen based on desired resolution, picked as 3 cm for the locate subsystem. The data for this selection appears in Figure A4-3. The other two frequencies exist at even logarithmic intervals between the very coarse and very fine frequencies.

The carbon epoxy casings may need to have small electric heaters inside of them, due to the sensitivity of SHIRAN components to very cold temperatures which exist on the moon. These casings also shield the SHIRAN components from cosmic radiation. As previously noted in this report, aluminum 6061-T6 can withstand the lunar environment, serving as a reliable antenna material.

2.6 Controls Subsystem

2.6.1. INTRODUCTION

Because of the degree of teleoperation needed for the system, an automated controls system is needed to accomplish certain tasks. The tasks of the controls subsystem are as follows:

- 1) To control the speed and direction of travel of the transport subsystem with respect to desired destination and feedback from the locate subsystem.
- 2) After directing the transport and load/unload subsystems to desired locations, the controls subsystem will control the opening and closing of the jaws of the modified Hayward bucket. The jaws need to open and close to transport ore, regolith and equipment (with mate).

2.6.2. RECOMMENDATION

TASK 1 - TRANSPORT ACTUATOR CONTROL

The controls system chosen to control the six actuators of the inverted Stewart platform is the process depicted in Figure 2-2 utilizing a hybrid / analog digital controller. The hybrid / analog controller consists of two major parts, namely a D/A position servo and a zero phase error tracking algorithm (ZPETA) prefilter (Lee, Yien). For the description of the control process, refer to Appendix 2-5. The actuators, motors themselves, will be Columbia H2000 AC Planetary Electric Hoists. These winches provide 1.5 kW of power and a 8900 N load rating. A ring composed of roller bearings will serve as a guide for the cable as it exits the winch. These winches will be mounted to the structure as determined by the structures team.

TASK 2- JAWS POSITION CONTROL

The control system to operate the jaws is a simple two - position control action. Though not drawn in Figure 2-2, the actuator will be attached in the same manner as the other actuators. The micro-computer in Figure 2-2 will compute a value which is a function of the four digital inputs received from the locate subsystem. When this value falls into predetermined and preprogrammed intervals, based upon initialization of locate subsystem, the output of the controller will switch. There are only two possible outputs for the controller, jaws open and jaws closed. A block diagram of this control system is given in Figure 2-3. The differential gap represents unintentional error, and the predetermined intervals discussed above. The actuator itself will be a 3 kW DC winch with a 1112.5 N load rating; it will be located inside the casing of the trolley attached to the jaws.

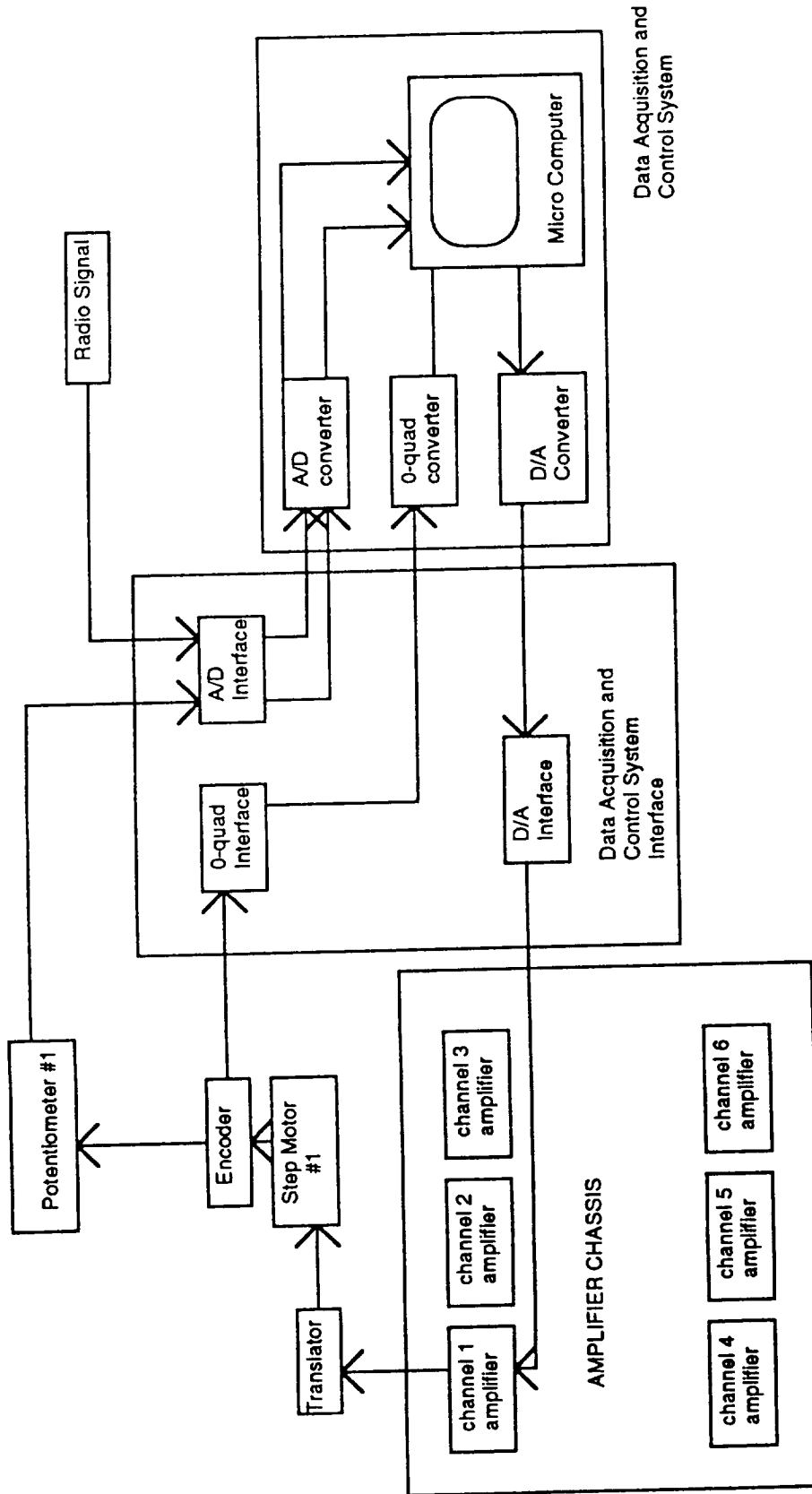


Figure 2-2 Control system architecture for the sturcture supported stewart platform actuators
for one of the six channels

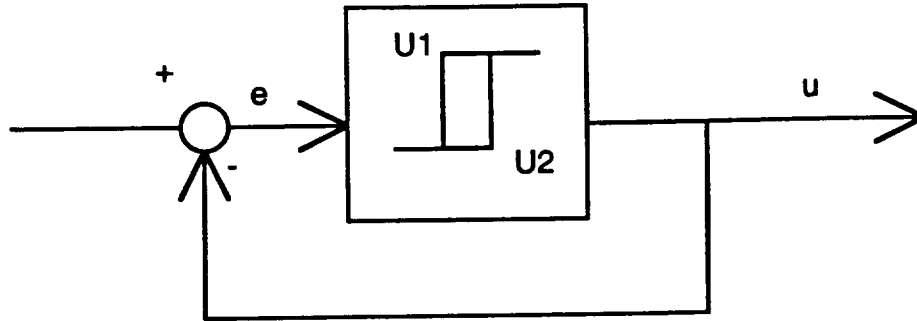


Figure 2-3: Two Position Control System

2.6.3. JUSTIFICATION

TASK 1

The hybrid analog digital controller will compensate for friction, which is increased on the moon, and eliminate error. The D/A position servo results in an effective non-linear friction compensation by shaping the static output characteristic of the actuator. The ZPETA achieves exact phase cancellation (zero error), (Lee, Yien). Hence the only error will be due to the signal resolution of the locate subsystem.

The Columbia H2000 AC Planetary Electric Hoist is the winch which NIST currently uses on their "Robo Crane," (Bostelman). Because the load conditions of the "Robo Crane" are comparable to the load conditions to be encountered by the transport subsystem, these winches were chosen. NIST uses these winches to lift roughly one metric ton of payload and that is the requirement of the transport subsystem. The ring of roller bearings is used to give the cable support and an extra degree of freedom that cannot be achieved with just the reel of the winch.

TASK 2

The two position controller was the obvious choice because of the simplicity of the controller is comparable to the requirements for this task. The controller is also inexpensive and widely used in industry. A 3 kW power requirement of the actuator is needed for the efficient operation of the jaws; the 1112.5 load rating

is needed because the jaws themselves weigh 890 N. The extra 222.5 N on the load rating is to account for momentum swings.

2.7 Interface with Mining Team

2.7.1. INTRODUCTION

The mining team will use the bottom platform (truss) of the transport subsystem for support during the mining operation. Because the load/unload subsystem (jaws) and the mining motor cannot occupy the bottom platform simultaneously, it is necessary to develop a way to interchange these two devices.

2.7.2. RECOMMENDATION

The solution chosen for this problem is the switchtrack subsystem (refer to Drawing 2-2 of Appendix 2-1). The subsystem will consist of an I-beam at an angle supported from above by structure fixed to the ground. The angle will be at 40 degrees above the horizontal to ensure rolling in lunar conditions. A two meter standard I-beam with a cross-sectional area as depicted in Figure 2-4 will be used. The support structure will consist of aluminum 6061-T6 pipes 5.1 cm (2") in outer diameter and 4.8 cm (1.75") inner diameter. The rest of the structure is composed of 0.16 cm, aluminum 6061-T6 cable. For a description of the interfacing process, refer to Appendix 2-6.

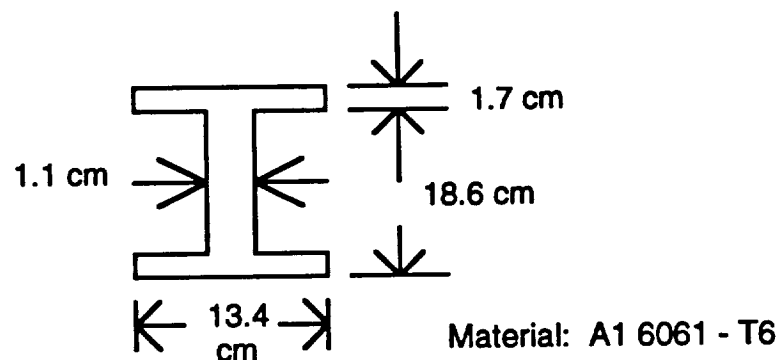


Figure 2-4: The Switchtrack I-Beam

2.7.3. JUSTIFICATION

The I-beam and trolley interface provides a stable and versatile interface between the load/unload subsystem and the transport subsystem and will be utilized. Therefore, the ability of the mining apparatus to interface with the truss in the same manner as the load/unload subsystem simplifies the interfacing between materials handling and mining. Below are the reasons for selecting the switchtrack as the method of interchange between the two subsystems.

- Requires no additional power source or power.
- Utilizes existing interface for load/unload subsystem
- Rugged and simple

The I-beam chosen was the same shape as that fastened on the bottom of the truss to assure proper mating. The support structure pipes and cable diameters were chosen based on the calculations in Appendix 2-6. The sheath allows for the 3 cm resolution capabilities of the locate subsystem and assures proper mating of the beams.

2.8 Dome Exit Interface

2.8.1. INTRODUCTION

Any ore recovered after mining must be delivered 5 meters outside the dome. The inverted Stewart platform can make it to the edge of the dome at best. Therefore, a method of transporting ore from the platform to 5 meters outside the dome is needed.

2.8.2. RECOMMENDATION

The chosen concept for this subsystem is a chute fastened to the dome structure at one end and suspended by cables at the other. The chute will be oriented at a 60 degree angle with the horizontal. It will extend into the dome 5 meters to allow the inverted Stewart platform to unload. The chute will be 3.2 meters wide at the unload point and 2 meters wide at the point of discharge from the structure. The cross-sectional profile will be U shaped, (see Figure 2-5). The

path to the 5 meter destination (transport vehicle) is completed by a free fall from the bottom of the chute. The chute thickness is 0.32 cm The material will be aluminum 6061-T6.

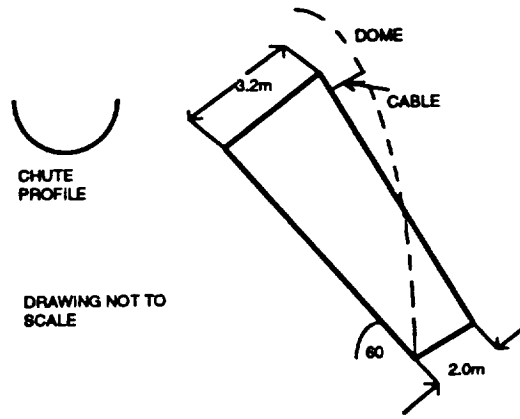


Figure 2-5: Chute Profile

2.8.3. JUSTIFICATION

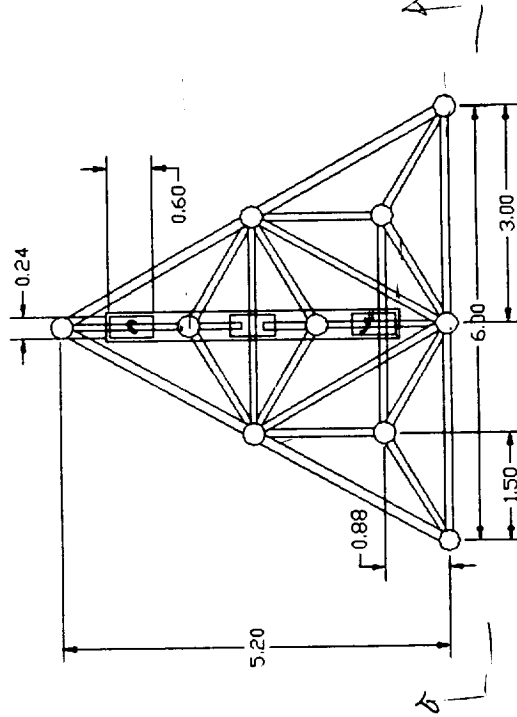
The chute widens at the top to account for spreading during unloading of the ore by the jaws. The dumping angle is maintained at 60 degrees, the optimum angle for bulk solids discharge from bins. The chute is narrowed at the bottom so that it may pass through the structural supports. This interface is a wise choice because it is very light, reliable and requires no power.

2.9 References

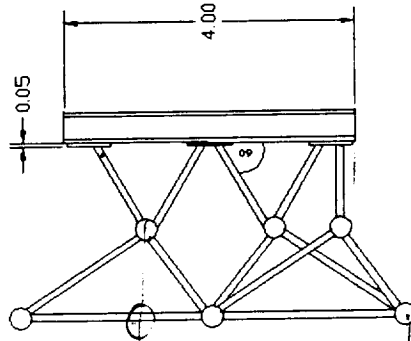
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NOTE: DRAWING NOT TO SCALE		MATERIALS HANDLING TEAM			
		TRANSPORT SYSTEM			
DRAWN	DATE	SIZE	TSCM NO.	DWG NO.	REV
BRAD WEAVER					
CHECKED					
R.D.					
DESIGN					
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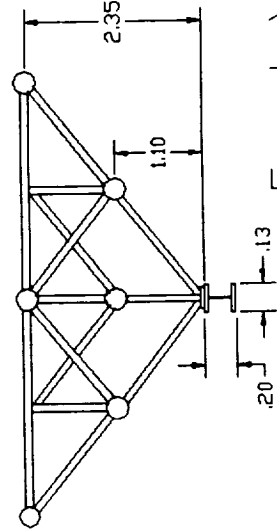
Top View



Side View

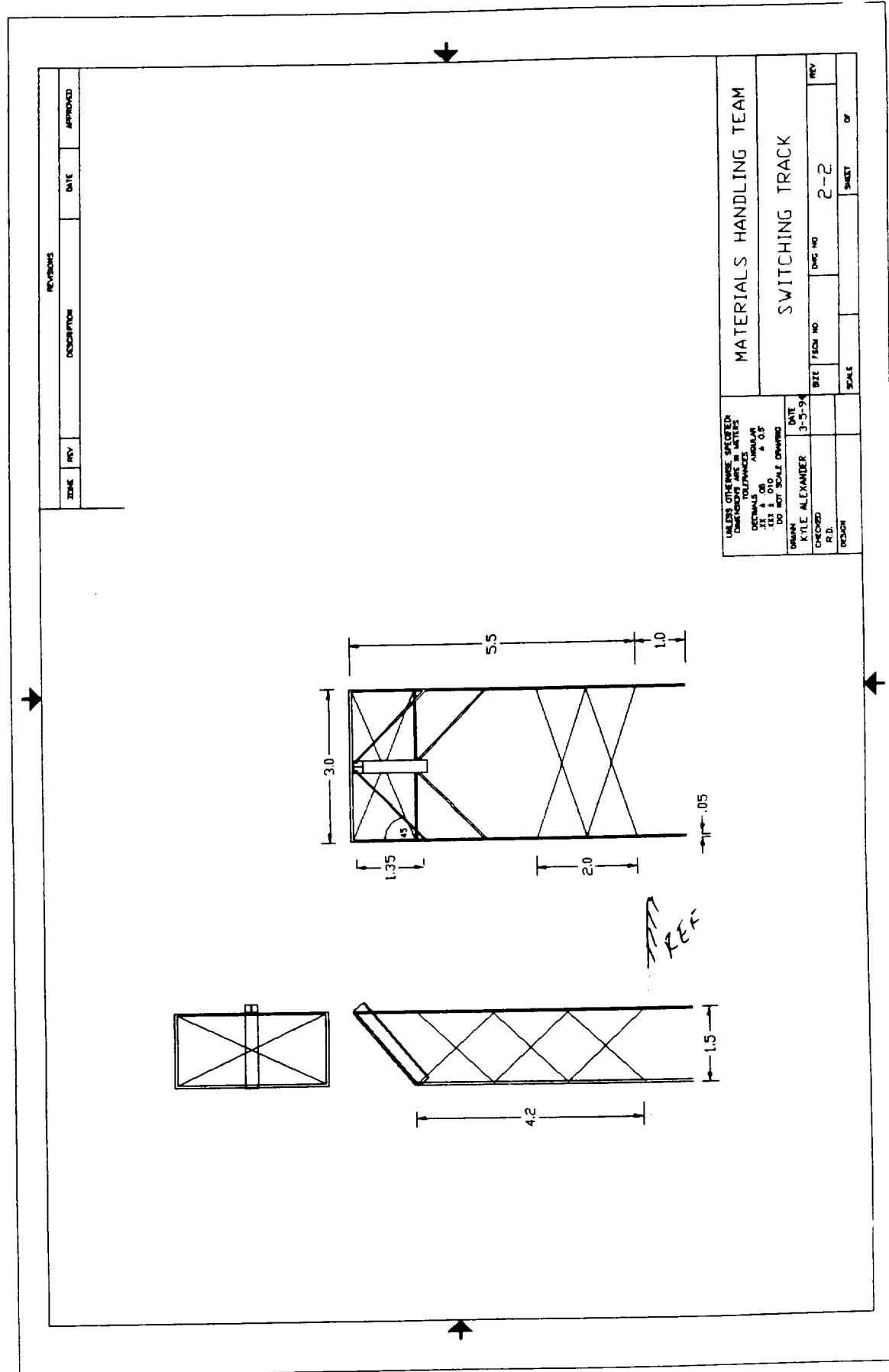


Front View



REVISIONS			
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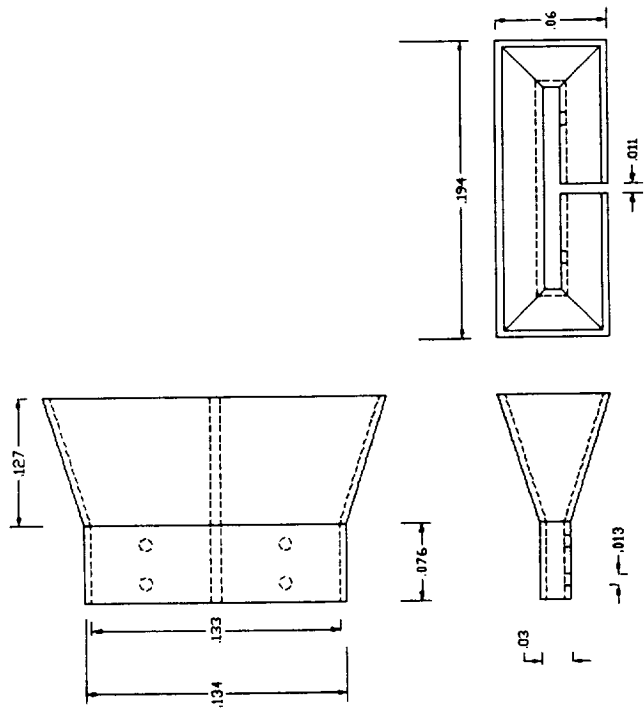
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		DWG NO.	OF
		2-1	



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ZONE	REV			



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MATERIALS HANDLING TEAM

SHEATH

Mat's Handling Group

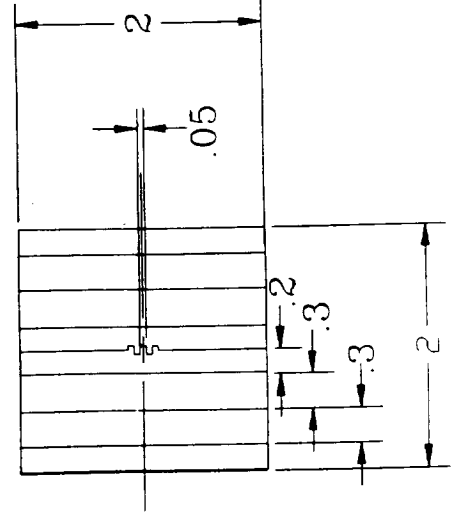
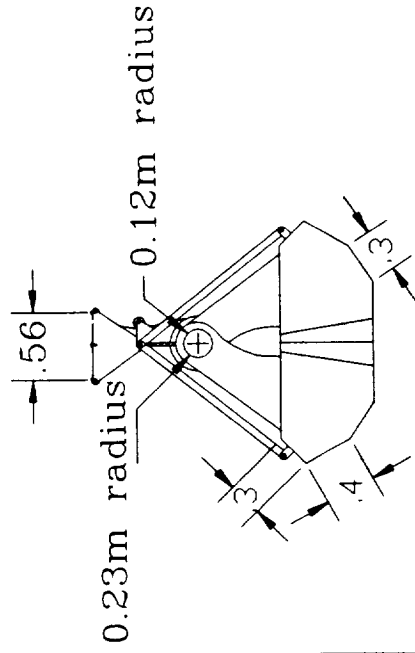
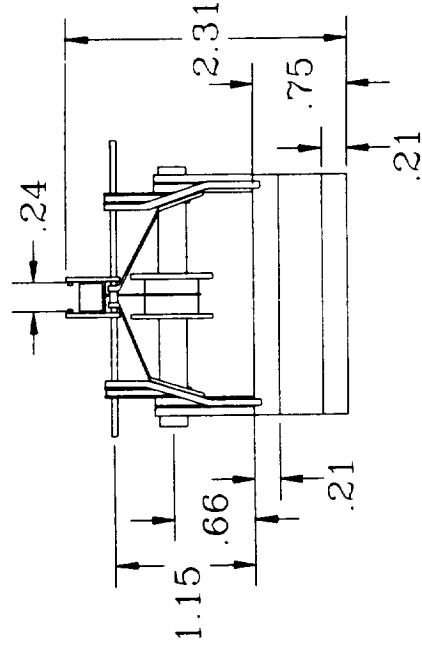
Title: Load/Unload Concept: The Jaws

Dwg #: MH-J1

Engr: R. Dailey

Date: 03/06/94

Rev #: 1-A



NOTE: ALL DIMENSIONS ARE IN METERS

APPENDIX 2-2: TRANSPORT SUBSYSTEM DESCRIPTION

RECOMMENDATION

The inverted Stewart Platform will best meet all necessary specifications. The inverted Stewart platform is a crane system which is composed of a stable triangular platform supported by six cables. It is capable of precisely positioning large loads in six degrees of freedom. It will be supported at three locations on the dome by six winches. The recommended material for the platform system is aluminum 6061-T6. The platform would consist of a modification involving the attachment of an I-beam along the base of the platform. The I-beam will serve as a track for the loading and unloading of equipment on and off of the platform. An additional modification involves building the platform as a truss structure as opposed to a solid triangular plate. The platform will weigh approximately 60 to 65 lbs on the moon and measure 6 meters to a side.

JUSTIFICATION

Concept: The cable supports along with the truss structure of the platform make this concept the lightest possible feasible solution. In addition, the cables remain in maximum stress a maximum amount of the time giving the system an optimum geometry. The I-beam allows for a simple method of transferring equipment and material and uses no power in the process. A representation of platform design is located in Appendix 2-1 (see Drawing 2-1).

Material of Construction: The recommended material of construction is aluminum 6061-T6. This material is described in section 2.2.

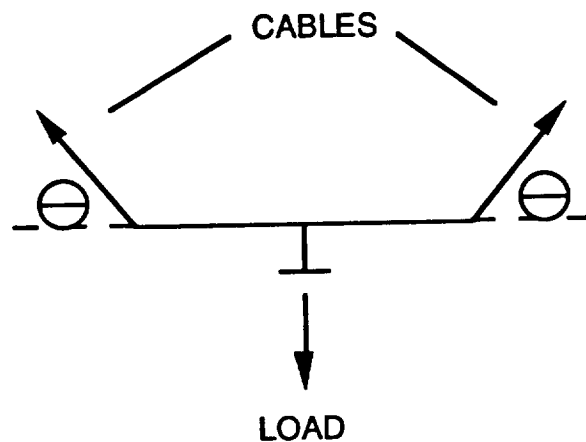
Analysis: The following areas are considered the most important considerations in designing the platform. Detailed calculations are included.

PLATFORM DESIGN

The platform will be supported at a distance of 20 m above the bottom of the structure. The three supports will be equally spaced 120° apart. Two winches

will reside at each support and will control one cable each. The cables will attach by ball and socket joints to the corners of the platform. Each corner of the platform will have two cables attached to it. This configuration gives the platform six degrees of freedom for movement within the structure. The platform itself is composed of trusses. An I-beam will be attached to the bottom of the platform. It will be secured by plates which are bolted to the beam and welded to the supporting truss members. The beam will serve as a track for the trolleys of the regolith jaws and the mining and drilling team's equipment. The track will also allow export of material outside of the structure.

The chosen size for each side of the platform is 6 m. It will be constructed of tubular beams of Aluminum 6061-T6. The maximum stress observed by the platform will be when the carried load on the trolley is positioned at the end of the I-beam as shown below:



The maximum axial loads on the beams occur when the two cables shown are in the same plane as their respective winches. As the angle between the cables and the platform decreases (as the platform is raised), the tension in the cables becomes increasingly greater. From the picture above, it can be seen that,

$$T = W / (2\sin\theta)$$

Therefore, a limit of 10° between the platform and its cables has been set. An angle of 10° in this position corresponds to a tension of approximately three times the load.

CABLES

Setting the maximum angle between the cables and the platform results in a maximum tension of approximately three times the hanging load. Therefore, the cables were designed to hold this load. The material chosen for the cables is Aluminum 6061-T6. The following equations were used to obtain a diameter for the cables:

$$\sigma_{ys} = P / A$$

where: P = total weight suspended
 σ_{ys} = yield stress of aluminum 6061-T6 (40 ksi)
 A = area of the cable

Therefore,

$$r = \sqrt{\frac{2P}{\pi\sigma_s}}$$

where: r = cable radius
 2 = safety factor

Using the calculated weights of the platform and jaws, and the given maximum weight from the drilling team, a cable diameter of 0.5 inches was calculated.

PLATFORM

The tubular beams which comprise the platform were also designed for the maximum axial loading. The material chosen for the beams is Aluminum 6061-T6. The following equations were used in designing the beams:

$$S.F. = \sigma_y / \sigma_{load}$$

where:

$$\sigma_{Load} = \frac{P_{Axial}}{A} = \frac{3W}{\pi(r_o^2 - r_i^2)}$$

Choosing an outer radius of 1.0 inches and inner radius of 0.875 inches yields a tubular beam with a safety factor of 4. The truss members which are welded to plates for attachment to the I-beam will have a different yield strength as a result of welding. These trusses will acquire the properties of Aluminum 6061-T6 which has a yield strength of 8 ksi. These members will have a inner radius of 0.50 inches and as a result will have a safety factor of over 2. A total of approximately 119 feet of beams are employed in the platform. The weight was calculated according to the equation:

$$\text{Weight} = \text{Volume} * \text{Sp. Wt. of Material}$$

where: sp. wt. of aluminum 6061-T6 = 170 lb/ft³

This weight was determined to be approximately 155 lbs on the earth.

The beam chosen for the platform is a standard 8x21 I-beam that is approximately 4 meters in length. The material chosen for the I-beam is Aluminum 6061-T6. Constructing a bending moment diagram, a maximum moment produced by the load of $WL/4$ can be found where L is the length of the beam. Therefore, the maximum stress due to bending is:

$$\sigma = My / I$$

The axial force in the beam was determined using the equation,

$$\sigma_{Load} = P_{Axial} / A$$

The total stress in the beam was therefore determined by the equation,

$$\sigma_T = \sigma_{Bending} + \sigma_{Axial} = (My / I) + (P / A)$$

where: $y = h / 2 = 4 \text{ in}$
 $I = 75.3 \text{ in}^4$

$$A = 6.16 \text{ in}^2.$$

The weight of the I-beam was calculated to be approximately 86 lbs on the earth.

RANGE OF MOTION

The inverted Stewart platform has a triangular range of motion. The winches for the platform will be mounted at a height of 20 meters above the base of the platform. The winches will be positioned 120° apart forming a triangle with sides of 19 meters. The resulting area coverage for the lower platform is 163 meters². This is approximately 23% of the total area of the base of the dome. The governing equations of motion have been done by NIST for the SPIDER (Albus). These equations apply to the materials handling platform. See Figure A2-1 for a graphical representation of the range of motion.

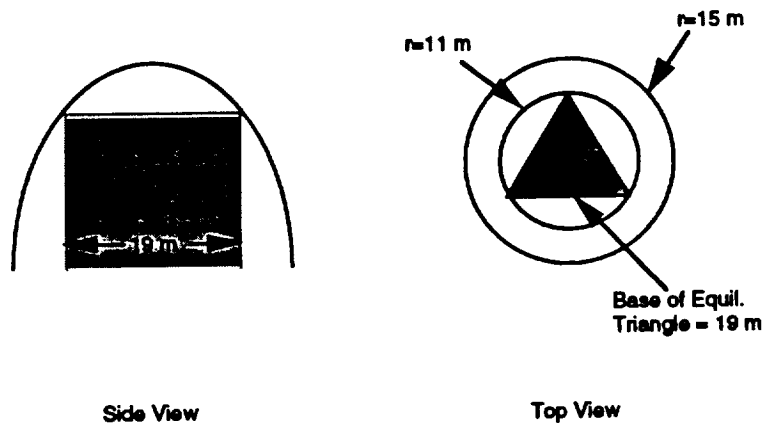


Figure A2-1

Appendix 2-3

Analysis of Jaws

Factor of Safety = 1.3

Given: Al6061-T6 (data from Mark's Handbook)

$E = 10.6E6 \text{ psi}$

$\nu = 0.33$

$\sigma_Y = 40kpsi$

$\sigma_U = 45kpsi$

Thermal Expansion = $13.5 \cdot 10^{-6} \text{ in} / (\text{in})^\circ F$

Thermal Conductivity = $90 \text{ Btu} / \text{hr} \cdot \text{ft} \cdot ^\circ F$

Specific Heat = $0.23 \text{ Btu} / \text{lb} \cdot ^\circ F$

App. $T_m = 1080^\circ F$

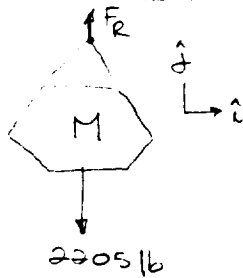
Elongation % = 17

Density = $2740 \text{ kg} / \text{m}^3$

Assumptions for Design of Shaft Mate:

- 1) Shaft and head are made of Al6061-T6
- 2) Forces on shaft act in pure tension
- 3) Radius between shaft and head = 0.1 in

ANALYSIS OF LOADING ON JAWS



- SEE MATERIAL PROPERTIES FOR A16061-T6
- ASSUME MOVEMENT OF $\ddot{y} = \left(\frac{32.2}{6}\right) \frac{ft}{s^2} \uparrow$

$$\left\{ \begin{array}{l} (M + 2205) \\ - \left(\frac{32.2}{6} \frac{ft}{s^2} \right) \end{array} \right\} \cdot \left(5.37 \frac{ft}{s^2} \right) = F_R - (M + 2205)$$

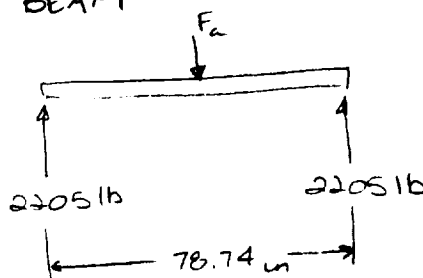
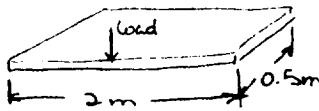
$$F_R = 2M + 4410$$

$$M = 200 \text{ lbs (SEE BELOW)}$$

$$F_R = 4810 \text{ lbs}$$

CALCULATE M1:

- WHAT IS WALL THICKNESS? TREAT JAW LIP AS CENTRALLY LOADED BEAM



$$F_R = 4410 \text{ lb}$$

$$M_a = 86,810 \text{ lb}\cdot\text{in}$$

$$\sigma = \frac{M}{I/c}$$

$$\sigma = \text{BENDING STRESS} = \frac{40 \text{ kpsi}}{1.2} = 30.8 \text{ kpsi}$$

$$M = M_a = 86,810 \text{ lb}\cdot\text{in}$$

$$I = \frac{bh^3}{12} = \frac{(78.74 \text{ in})h^3}{12}$$

$$c = h/2$$

$$30.8 \text{ kpsi} = \frac{86,810 \text{ lb}\cdot\text{in}}{\left(\frac{78.74 \text{ in} h^3}{12} \right) \left(\frac{h}{2} \right)}$$

$$\frac{(30.8 \text{ kpsi})(78.74 \text{ in})(2)}{12(86,810 \text{ lb}\cdot\text{in})} = \frac{1}{h^2} \Rightarrow h = 0.46 \text{ in}$$

$$\text{WE WILL USE } h = 0.5 \text{ in}$$

$$\text{VOLUME}^* = 2 \left\{ [0.5 \text{ m} + 0.3 \text{ m} + 0.4 \text{ m} + 0.3 \text{ m} + 0.8 \text{ m}] (0.0127 \text{ m} \times 2 \text{ m}) + (0.8 \text{ m})^2 (0.0127 \text{ m}) \right. \\ \left. + (0.5 \text{ m} \times 0.75 \text{ m}) + \frac{1}{2} (0.75 \text{ m} \times 0.3 \text{ m}) + \frac{1}{2} (0.1 \text{ m} \times 0.275 \text{ m}) + \frac{1}{2} (0.31 \text{ m} \times 0.2 \text{ m}) \right. \\ \left. + \frac{1}{2} (0.275 \text{ m} \times 0.4 \text{ m}) + \frac{1}{2} (0.4 \text{ m} \times 0.2 \text{ m}) \right\} (0.0127 \text{ m}) = 0.15 \text{ m}^3$$

$$\text{DENSITY (A16061-T6)} = 2740 \text{ kg/m}^3$$

$$\text{Mass}^* = 408.3 \text{ kg} \approx 900 \text{ lbs (earth)} \\ = 150 \text{ lbs (lunar)}$$

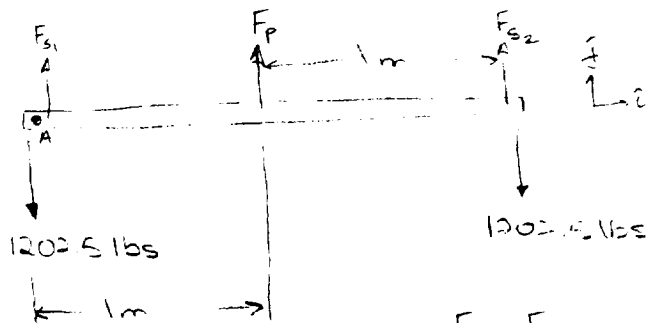
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* DOES NOT INCLUDE DRUMS, CHAIN,
ROPE OR TROLLEY SET-UP

ASSUME THIS WT \approx 50 lb (LUNAR)

FOR TOTAL WT: $12 \times 2 \text{ lbs (chain)} + 50 \text{ lbs}$

ANALYSIS OF SMALL DRUM



WANT ACCELERATION = $0.5 \frac{F_1}{F_2}$
 $= 6 \text{ in/sec}^2$

$$F_{s1} = F_{s2}$$

$$F_s = F_{s1} + F_{s2}$$

$$F_p + F_s - 2405 = \left(\frac{2405}{g} \right) (0.5 \frac{F_1}{F_2})$$

$$F_p + F_s = 2629$$

$$\sum \tau_A: (39.37 \text{ in}) F_p + (78.74 \text{ in}) \left(\frac{1}{2} F_s \right) - (78.74 \text{ in}) (1202.5 \text{ lbs}) = J \ddot{\theta}$$

$$J = \frac{1}{2} M r^2 = \left(\frac{1}{2} \right) \pi r_s^2 (78.74 \text{ in}) \left(3.08 \cdot 10^{-3} \frac{\text{slugs}}{\text{in}^3} \right) r_s^2$$

$$= 0.381 \text{ slug} \cdot \text{in}^4$$

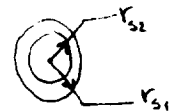
$$\omega = \dot{\theta} / r_s = g / r_s \text{ in our case}$$

$$F_p + F_s = 2.29 r_s^3 + 2405$$

$$2629 = 2.29 r_s^3 + 2405$$

$$r_s = 4.60 \text{ in}$$

BENDING: (ASSUME $F_{s1} = F_{s2} = 0$) HOLLOW TUBE



$$\sigma = \frac{M}{(I/c)} \Rightarrow M = (39.37 \text{ in}) (1202.5 \text{ lb}) = 43,343 \text{ in} \cdot \text{lb}$$

$$(I/c) = \frac{\pi}{4} \left(\frac{(4.60 \text{ in})^4 - r_{s2}^4}{4.60 \text{ in}} \right)$$

$$\sigma = \sigma_r = \frac{40 \text{ kpsi}}{1.3} = \frac{43,343 \text{ in} \cdot \text{lb}}{\frac{\pi}{4} \left(\frac{(4.60 \text{ in})^4 - r_{s2}^4}{4.60 \text{ in}} \right)} \Rightarrow r_{s2} = 4.58 \text{ in}$$

WE SHALL USE $4.1 \text{ in} = r_{s2}$

WEIGHT OF SMALL DRUM:

$$W = \rho V$$

$$\rho = 0.099 \frac{\text{lbs}}{\text{in}^3} \text{ on EARTH}$$

$$V = \pi (r_o^2 - r_{s2}^2) = \pi ((4.6 \text{ in})^2 - (4.1 \text{ in})^2) (78.74 \text{ in}) = 1076.1 \text{ in}^3$$

$$\text{WT OF SMALL DRUM} = 17.8 \text{ lbs (on Moon)}$$

DRUM SIZES (POWER REQUIREMENT)

$$F_R = 2405 \text{ lbf} = \text{REQUIRED FORCE TO LIFT JAW} = 2205 \text{ lbf}$$

ASSUME WE WANT TO MOVE MASS AT ACCELERATION OF $20.5 \frac{\text{ft}}{\text{s}^2}$ FOR INDEFINITE TIME

ASSUME WE WANT TOP VELOCITY OF $1 \text{ ft/s} = 12 \text{ in/sec}$

$V = Gt \text{ (in/sec)}$
THIS IS MUCH LESS THAN WHAT THE JAW WAS DESIGNED FOR (537 ft/s → SO JAW CAN DROP OR FALL)

P_R = POWER REQUIREMENT

T_R = REQUIRED TORQUE

ω_s = ANGULAR VELOCITY OF SMALL D.C.

r_s = RADIUS OF SMALL DRUM

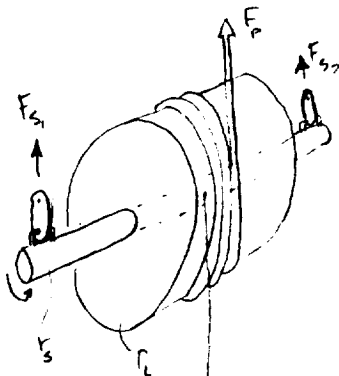
V = UPWARD VELOCITY OF EITHER DRUM

ω_L = ANGULAR VELOCITY OF LARGE DRUM

r_L = RADIUS OF LARGE DRUM

V_L = UPWARD VELOCITY OF LARGE DRUM

F_P = REQUIRED PULLING FORCE



RELATIONS: $V = \omega_s r_s$

$$T_R = r_L F_P$$

$$\omega_s = \omega_L$$

$$P_R = T_R \omega_L$$

$$\therefore V = \omega_L r_s \rightarrow \omega_L = V/r_s$$

$$\sum F_y: F_s + F_P - F_R = \left(\frac{F_R}{g}\right) \left(0.5 \frac{\text{ft}}{\text{s}^2}\right) \quad F_s + F_P = 1.09 F_R = 2629 \text{ lbf} = F_s + F_P$$

$$\sum M: -F_s r_s + F_P r_L = J \dot{\omega} \quad \dot{\omega} = V/r_s = \frac{0.5 \frac{\text{ft}}{\text{s}^2}}{r_s} = \frac{6 \text{ in/s}^2}{r_s}$$

$$J = \frac{1}{2} M_s r_s^2 + \frac{1}{2} M_L r_L^2$$

$$M_s = (\pi r_s^2) (78.74 \text{ in}) (3.08 \cdot 10^{-3} \frac{\text{slugs}}{\text{in}^3}) = 0.762 r_s^2$$

$$M_L = (\pi r_L^2) (19.69 \text{ in}) (3.08 \cdot 10^{-3} \frac{\text{slugs}}{\text{in}^3}) = 0.191 r_L^2$$

$$J = 0.762 r_s^2 + 0.191 r_L^2$$

$$J \dot{\omega} = (0.762 r_s^2 + 0.191 r_L^2) \left(\frac{6}{r_s}\right) = -F_s r_s + F_P r_L$$

$$P_R = T_R \omega_L = F_P r_L \omega_L = F_P r_L (V/r_s) = \left(\frac{r_L}{r_s}\right) F_P (Gt) = P_R$$

$$F_s = 2629 - F_P$$

$$-(2629 - F_P) r_s + F_P r_L = (0.762 r_s^2 + 0.191 r_L^2) \left(\frac{6}{r_s}\right)$$

$$-2629 r_s + (F_P \times r_L + r_s) = (0.762 r_s^2 + 0.191 r_L^2) \left(\frac{6}{r_s}\right)$$

$$F_P = \frac{2629 r_s + (0.762 r_s^2 + 0.191 r_L^2) \left(\frac{6}{r_s}\right)}{(r_s + r_L)}$$

DRUM SIZES (POWER REQUIREMENT) - (CONT)

WITH $r_s = 4.6$ in

$$F_p = \frac{2629(4.6) + (0.7162(4.6)^4 + 0.191r_L^4)(4.6)}{4.6 + r_L} = \frac{12434.6 + 0.25r_L^4}{4.6 + r_L}$$

$$F_p = \frac{12434.6 + \frac{1}{4}r_L^4}{4.6 + r_L}$$

$$P_R(t) = \left(\frac{r_L}{4.6}\right)(\omega_L) \left(\frac{12434.6 + 0.25r_L^4}{4.6 + r_L}\right)$$

WE WANT A SMALL P_R , THUS A LARGE r_L

r_L IS RESTRICTED IN SIZE
BY CLEARANCE, r_s

$$4.6 \text{ in} < r_L < 22.3 \text{ in}$$

$$P_R < 25 \text{ kW}$$

ASSUME TOP VELOCITY OF 12 in/sec, OR $T = 2$ sec

$$P_R(2) = \left(\frac{r_L}{4.6 + r_L}\right)(0.652r_L^4 + 32438.0)$$

$$P_R < 25,000 \text{ W} = 33.53 \text{ hp} = 18439.05 \frac{\text{ft} \cdot \text{lb}}{\text{sec}} = 221268 \frac{\text{in} \cdot \text{lb}}{\text{sec}}$$

$$221,268 = \left(\frac{r_L}{4.6 + r_L}\right)(0.652r_L^4 + 32,438.0)$$

$r_L = 24.4 \text{ in} > 22.3 \text{ in} \rightarrow$ SO THE MAX R IS OUTSIDE
OF OUR RANGE ANYWAY

RECALL $F_p + F_s = 2629$ (ie, $F_s = 2629 - F_p$) & $F_p \leq F_s$ (SINCE $r_p \geq r_s$)
OR $F_p \leq \frac{2629}{2}$

FOR $F_p = 1315 \text{ lb}$, $r_L = 15.3 \text{ in}$

$F_p = 657.5 \text{ lb}$, $r_L = 10.1 \text{ in}$

$F_p = 0$

BASED ON GRAPH, $r_L = 9.25$ WILL
GIVE BEST COMPROMISE

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$$\begin{aligned} r_s &= 4.6'' \\ r_L &= 9.25'' \end{aligned}$$

$$F_p = 1029.95 \text{ lb}$$

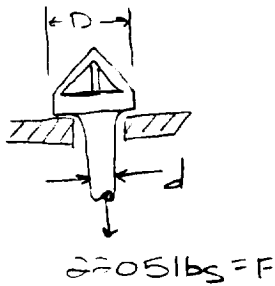
$$P_R = 24852.28 \frac{\text{in} \cdot \text{lb}}{\text{sec}} = \boxed{2.8 \text{ kW}}$$

$$Wt_s = 17.8 \text{ lbs}$$

$Wt_L < 87.7 \text{ lbs}$ (SOLID Wt) \rightarrow (10 WITH DRUM THAT
HAS INNER SUPPORTS

STATIC DESIGN FOR JAW MATE (SEE MAT'L PROPERTIES FOR AL6061-T6)

factor of safety = 1.3



$$\sigma_D = \frac{2205 \text{ lbs} (1.3)}{\frac{\pi}{4} (D^2 - d^2)}$$

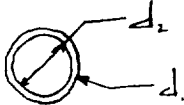
$$\sigma_d = \frac{2205 \text{ lbs} (1.3)}{\frac{\pi}{4} (d_1^2 - d_2^2)}$$

$$\sigma_T = 40 \text{ kpsi} = \frac{2205 \text{ lbs} (1.3)}{\frac{\pi}{4} (d_1^2 - d_2^2)}$$

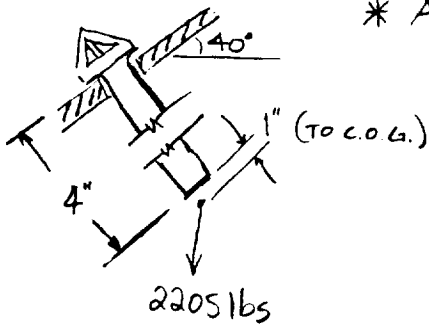
$$(d_1^2 - d_2^2) = 9.12 \cdot 10^{-2} \text{ in}^2$$

SHAFT IS HOLLOW TUBE

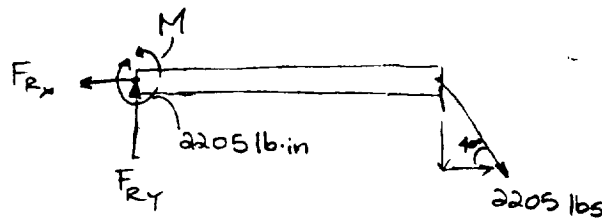
$d \Rightarrow$



$$\sigma_d = \frac{\sigma_T}{\frac{\pi}{4} (d_1^2 - d_2^2)}$$



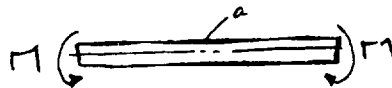
* ARBITRARILY SET LENGTH OF SHAFT @ 4"



$$F_{Rx} = 2205 \cos 40^\circ = 1689.13 \text{ lbs}$$

$$F_{Ry} = 2205 \sin 40^\circ = 1417.35 \text{ lbs}$$

$$M = 2205 \text{ lb} \cdot \text{in} + 4 \text{ in} (1417.35 \text{ lb}) = 7874.39 \text{ lb} \cdot \text{in}$$



WHERE $M = 7874.39 \text{ lb} \cdot \text{in}$
 $n = 1.3$

$$\sigma = M / (I/c)$$

$$I/c = \frac{\pi}{4} \frac{(d_1^4 - d_2^4)}{d_1}$$

ALLOWABLE BENDING STRESS, $\sigma = \frac{40 \text{ kpsi}}{1.3} = 30.8 \text{ kpsi}$

$$\frac{\pi}{4} \frac{(d_1^4 - d_2^4)}{d_1} = \frac{7874.39 \text{ lb} \cdot \text{in}}{30,800 \text{ lb/in}^2} \rightarrow \left(\frac{d_1^4 - d_2^4}{d_1} \right) = 0.326 \text{ in}^3$$

STATIC DESIGN OF JAW PLATE (CONT.)

$$\frac{d_1^4 - d_2^4}{d_1} = 0.326 \text{ in}^3$$

$$(d_1^2 - d_2^2) = 9.12 \cdot 10^{-2} \text{ in}^2 \rightarrow d_2^2 = d_1^2 - 0.0912 \text{ in}^2$$

$$d_1^4 - (d_1^2 - 0.0912 \text{ in}^2)^2 = 0.326 d_1$$

$$d_1^4 - d_1^4 + 0.1824 d_1^2 - 0.00832 \text{ in}^4 - 0.326 d_1 = 0$$

$$0.1824 d_1^2 - 0.326 d_1 - 0.00832 \text{ in}^4 = 0$$

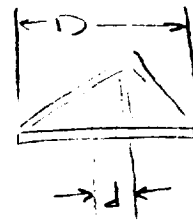
$$d_1 = \frac{0.326 \pm \sqrt{0.326^2 - 4(0.1824)(-0.00832)}}{2(0.1824)}$$

$$\boxed{d_1 = 1.81 \text{ in}}$$

$d_1 = 1.61 \text{ in}$ due to fatigue

$$\boxed{d_2 = 1.79 \text{ in}}$$

thickness = 0.02 in



CALCULATE HEAD DIAMETER

$$\sigma_F = \frac{(1.3)(2205 \text{ lb})}{\frac{\pi}{4}(D^2 - d^2)}$$

$$d = d_1 = 1.81 \text{ in}$$

$$\sigma_F = \sigma_T = 40 \text{ kpsi}$$

$$\left\{ \left(\frac{\pi}{4} (40 \text{ kpsi}) \right) \right\}^{-1} + (1.81)^2 = D^2 \Rightarrow D = 1.84 \text{ in}$$

→ THAT ONLY GIVES
 $\frac{1}{2}(1.84 \text{ in} - 1.81 \text{ in}) = 0.015 \text{ in}$
 FOR JAWS TO HOLD PER
 SIDE. WE WANT THE
 JAWS TO HAVE AT LEAST
 AN INCH TO GRAB, SO
 MAKE $\boxed{D = 3.81 \text{ in}}$

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FATIGUE ANALYSIS OF JAW PLATE

SODERBERG: $\frac{n\sigma_a}{S_e} + \frac{n\sigma_m}{S_T} = 1$ $n \approx 1.3$

$S_e = 14 \text{ kpsi}$ (FROM FLACK'S HANDBOOK.)

$S_T = 40 \text{ kpsi}$

$\sigma_a = \frac{4810 \text{ lbs}}{\frac{\pi}{4}(1.81^2 - 1.79^2) \text{ in}^2} = 85.1 \text{ k lb/in}^2$

$\sigma_m = \frac{4810 \text{ lbs} (\frac{1}{2})}{\frac{\pi}{4}(1.81^2 - 1.79^2) \text{ in}^2} = 42.5 \text{ k lb/in}^2$

$\frac{85.1 \text{ kpsi}}{14 \text{ kpsi}} + \frac{42.5 \text{ kpsi}}{40 \text{ kpsi}} = 7.14 = \frac{1}{n}$ $n = 0.14 \rightarrow \text{no good}$
MAKE ID SMALLER

$\left(\frac{\frac{4810}{\frac{\pi}{4}(1.81^2 - d_2^2)}}{14 \text{ k}} \right) + \left(\frac{\frac{2405}{\frac{\pi}{4}(1.81^2 - d_2^2)}}{40 \text{ k}} \right) = \frac{1}{1.3}$

$\frac{(40 \text{ k})(4810)}{(1.81^2 - d_2^2)} + \frac{(14 \text{ k})(2405)}{(1.81^2 - d_2^2)} = \left(\frac{1}{1.3} \right) (40 \text{ k})(14 \text{ k}) \left(\frac{\pi}{4} \right)$

$(1.81^2 - d_2^2) = 0.668$

$\boxed{d_2 = 1.61 \text{ in}}$

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APPENDIX 2-4: LOCATE SUBSYSTEM DESCRIPTION

The scientific basis and modeling of this system are based on the fact that as radio waves travel through free space (a theoretical vacuum), they exhibit a phase shift, or phase delay, directly proportional to distance (Cubic). A data converter calculates "range data" (the distance to each beacon) from measured phase delays. The calculated distances serve as four coordinates that outline the position of any device attached to the platform. These values are the input for the Controls Subsystem. As the materials handling unit moves items around the crater, computer memory stores the current location of all objects in the crater.

The SHIRAN locate subsystem initializes when it receives a coded 20 megahertz (MHz) initialization radio signal from the controls subsystem. The main transceiver, mounted to the transport subsystem, then transmits another coded 20 MHz signal that turns on one of the beacons, which is actually a transceiver itself. These coded signals are radio transmissions having characteristic bursts of differing duration, similar to Morse code. Each beacon has its own unique initialization signal. An entire coded signal will last for only 20 milliseconds (ms). The main transceiver then transmits a ranging signal to the initialized beacon. The beacon retransmits the signal back to the main transceiver. This sequence is conducted four times for each beacon, since there are four ranging frequencies (to avoid ambiguity of the data). Each of the four sequences consumes 10 ms (Cubic). An electronic phasemeter measures the phase delay by comparing the original ranging transmission to the received, retransmitted signal. A data converter then converts the measured phase delays into "range data", the distance to the beacon. The range data is then transmitted to the controls subsystem via another coded 20 MHz signal, a process taking about 25 ms. Range data is digital input to the controls subsystem. This process is then repeated for the other three beacons. For a visual representation of this process, please see Figure A4-1 below.

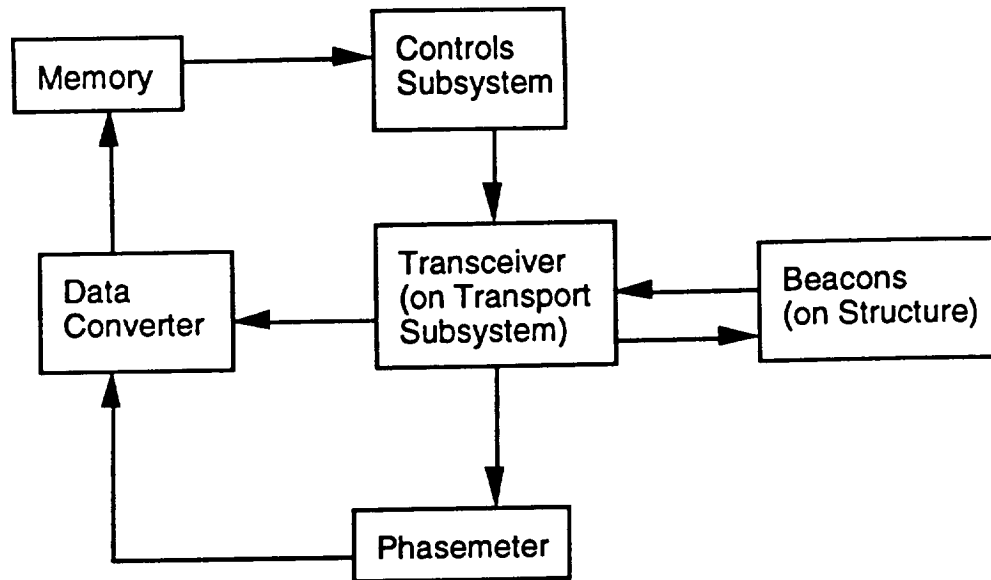


Figure A4-1

The total amount of time required to obtain a complete set of range data is quite small.

$$\text{Time to measure phase delays} = (20 + [10 \times 4] + 25 \text{ ms}) \times 4 \text{ beacons} = 340 \text{ ms}$$

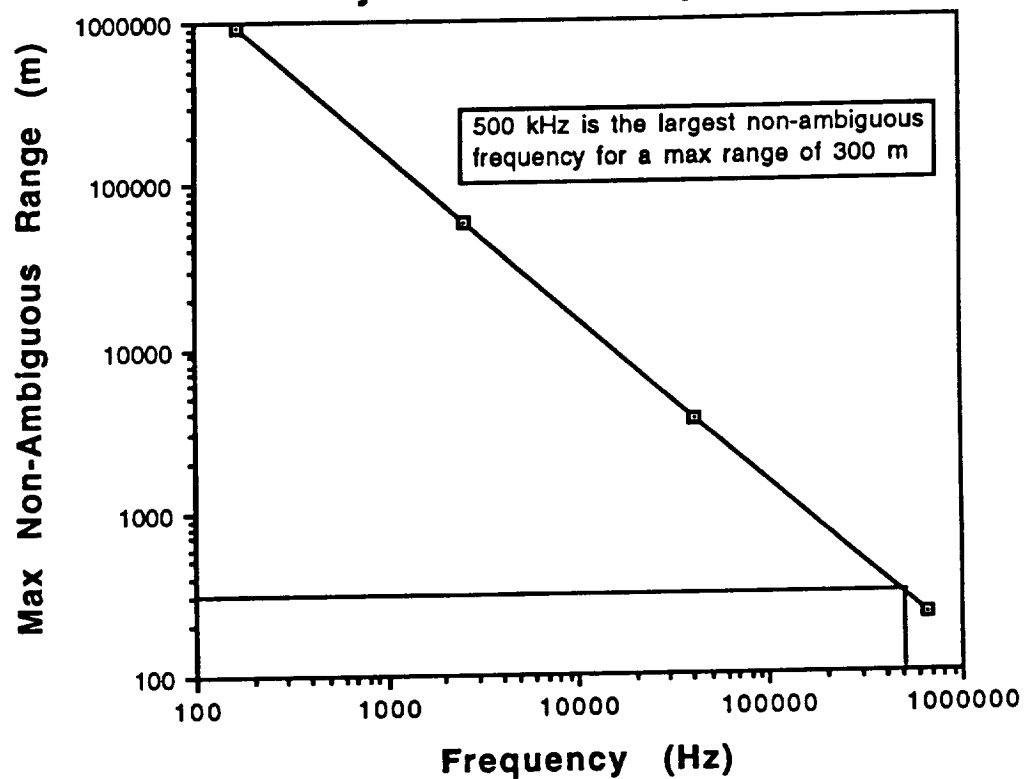
Assuming that the data conversion process consumes about 100 ms, the SHIRAN system is able to update the position of the transport subsystem two times per second. The transceiver mounted on the transport subsystem has a fixed position in reference to the load/unload subsystem. When loading and unloading occur, the data converter can adjust the range data accordingly.

All four beacons will be placed on the structure in the horizontal plane at the same height as the transport subsystem's winches. This plane is circular, with one beacon being at its center. The other three beacons are placed at 120 degree intervals along the circumference of the circle, directly opposing the winches.

The selection of the four ranging frequencies was based on previously published data (Cubic). The frequencies are described by the following titles: very coarse, coarse, fine, and very fine. One must select the very coarse frequency so that

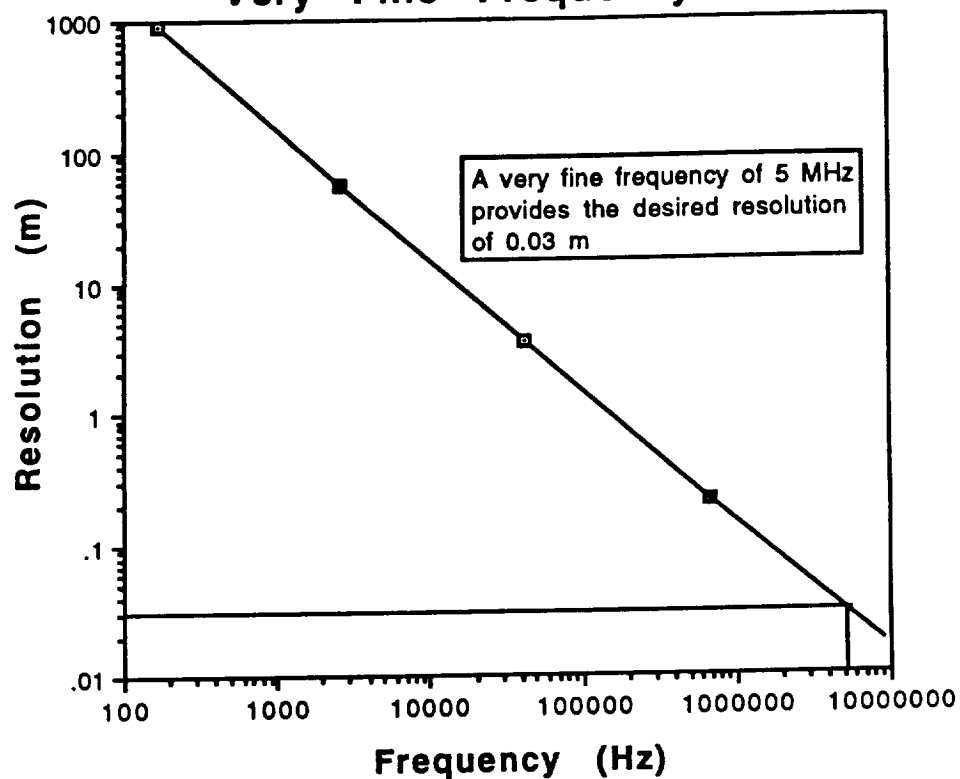
the maximum phase delay corresponds to the maximum distance to be measured. In the lunar mining scenario, this will be at most 300 meters. The very coarse frequency selection arises from the data shown in Figure A4-2.

Figure A4-2: Selection of Very Coarse Frequency



The very fine frequency must be chosen based on desired resolution, picked as 3 cm for the locate subsystem. The data for this selection appears in Figure A4-3.

Figure A4-3: Selection of Very Fine Frequency



The other two frequencies exist at even logarithmic intervals between the very coarse and very fine frequencies. Therefore, the four ranging frequencies are as follows:

Very Coarse = 500 kHz

Coarse = 1.08 MHz

Fine = 2.33 MHz

Very Fine = 5.00 MHz

The initial setup of the lunar mining project requires that mining and drilling equipment be placed in the crater before the construction of the structure. This poses a problem for the locate subsystem. Upon initialization, the materials handling system must know the location of these items. After the beacons are attached to the structure, a manned team can carry the main transceiver (before its attachment to the inverted Stewart platform) into the crater to find out exactly where these items are in reference to the beacons. These teams can also check

and calibrate the system using surveying devices in conjunction with the transceiver.

In previous use, SHIRAN components have shown a very low tolerance to temperatures below -20 degrees Celsius, while having no known problems at any higher temperatures (Cubic). It is recommended that the beacons and transceiver be placed in carbon epoxy casings with a small electric heater inside, to counteract the very cold temperatures that occur on the moon. The casings also need to shield the components from cosmic radiation, a condition unencountered by any other SHIRAN system. The antennas on the transceiver and beacons are bow tie antennas, and will need to be made of aluminum 6061-T6, which can withstand the lunar environment.

APPENDIX 2-5: CONTROL PROCESS

Radio signals from the locate subsystem are fed into an analog-to-digital (A/D) board embedded in the computer, refer to Figure 2-2 in Section 2.6. The output is 4 digital numbers specifying the desired position. From these numbers, the computer calculates the desired position and orientation of the mobile platform and translates it into actuator cable lengths. Feedback is received from cable length and force sensors located near the winches (step motors). This enables closed loop position, velocity, and force control. Cable travel encoders generate phase quadrature signals for the encoder input board embedded in the computer. Cable tension sensors are input into an A/D board in the computer. Command signals are output from the computer via a digital to analog (D/A) board and sent to the winch amplifiers,(Albus, Bostelman, Daglakis, 1992). The controller used in the micro-computer will involve a hybrid / analog digital controller. The hybrid / analog controller consists of two major parts, namely a D/A position servo and a ZPETA prefilter (Lee, Yien).

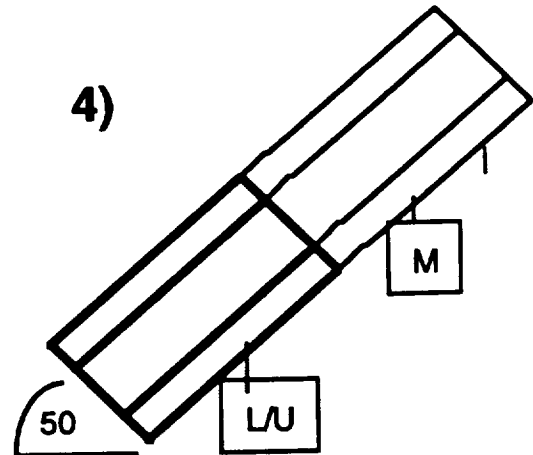
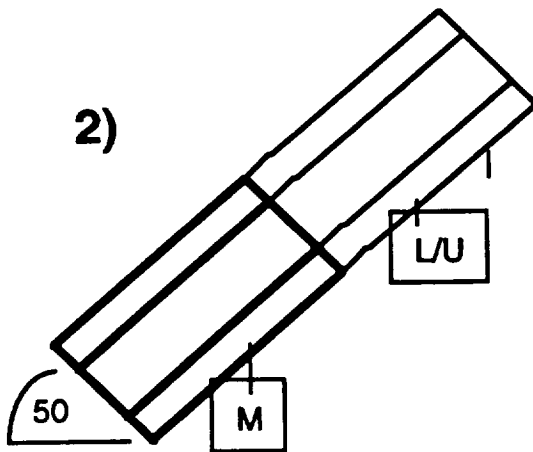
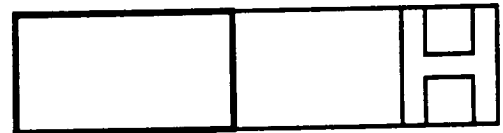
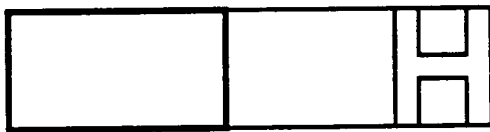
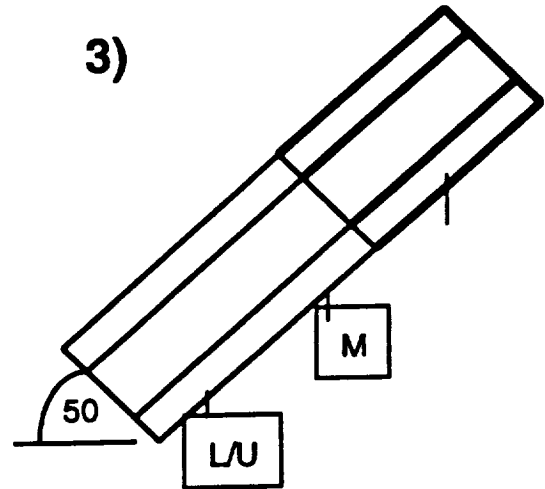
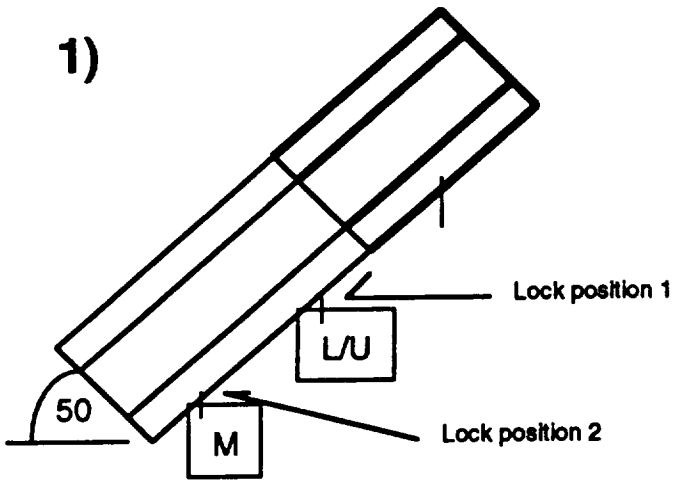
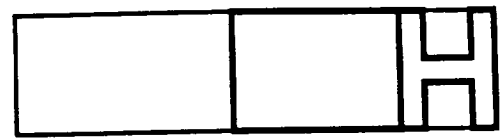
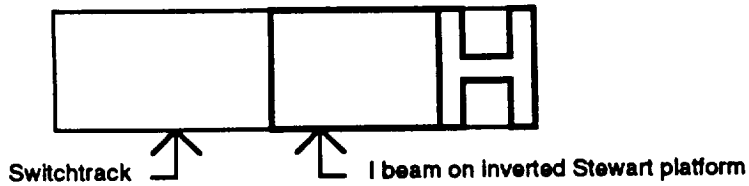
APPENDIX 2-6: MINING EXCHANGE PROCESS

Refer to Figure A6-1.

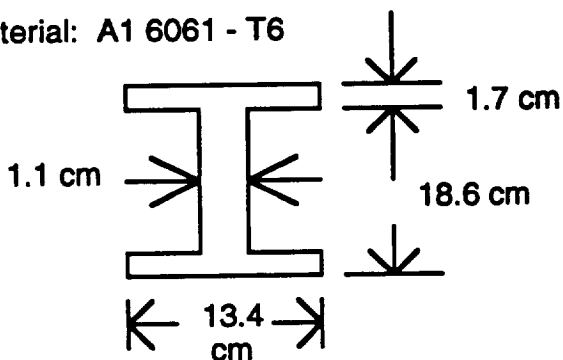
Initially, the load / unload (L/U) system is on the inverted Stewart platform while mining awaits at lock position 2 on the I beam.

- 1) Platform aligns with I beam, L/U is released from platform and rolls on to I beam and locks at position 1 on I beam via mechanical locks
- 2) Platform aligns with I beam on other side near Mining subsystem (M) Lock at position 2 on I beam releases, Mining rolls on to platform and locks into place. As platform disengages, it triggers release of lock at position 1 on I beam, and L/U rolls until it locks at position 2 on I beam.
- 3) After mining is completed, platform aligns with I beam as it did in STEP 1. Mining is released from platform and rolls on to I beam and locks at position 1 on I beam.
- 4) Platform aligns with I beam as it did in STEP 2. Lock at position 2 on I beam releases, L/U rolls on to platform and locks into place. As platform disengages, it triggers release of lock at position 1 on I beam, and Mining rolls until it locks at position 2 on I beam. The cycle is completed.

The mating of the two I beams will be facilitated by the "Sheath". Refer to Drawing 2-3 of Appendix 2-1. A "sheath" will be connected to the top flange of each end of the I beam. The inside of this sheath is lined with roller bearings to decrease friction during alignment. Note that the sheath is bolted to the switch track I beam with the same bolts used to fasten the 45 degree angle bars of the supporting structure. One sheath will be placed on each end of the I beam.



Material: A1 6061 - T6



THICK LINED I BEAM REPRESENTS I BEAM ON INVERTED STEWART PLATFORM

NORMAL LINES ARE THE SWITCHTRACK

LU = LOAD / UNLOAD SUBSYSTEM

M = MINING APPARATUS

Figure A6-1

DESIGN AGAINST BENDING AND BUCKLING CALCULATIONS FOR SWITCHTRACK

Worst case bending is represented by one bar member supporting the I beam and the jaws. The jaws weigh 200 lbs on the moon and the I beam will weigh 140 lbs. Thus, the worst case is one 3 meter section of pipe supporting 340 lbs in the middle

This corresponds to a maximum moment of 1674 ft lbs.

The yield stress for A1 6061-T6 is 40 ksi.

$$\sigma = \frac{My}{I} \quad (1)$$

The above equation defines the maximum bending stress. For this situation, y = the outer radius of the pipe, I is the moment of inertia.

$$I = 2\pi r^3 t \quad (2)$$

t is the thickness of the pipe and r is the outer radius. Substituting into equation 1, an equation in terms of the thickness and radius is available. Choosing a value for r_2 yields the minimum thickness of the pipe. Note that a safety factor of 2 has been included.

$$\sigma = \frac{Mr_2}{2\pi r^2 t}$$

$$t = \frac{Mr_2}{2\pi r^2 (20ksi)}$$

With r_2 set at 2 inches, the minimum thickness defined by the above equation is .04 inches, or smaller than 1/16". This thickness is not intuitively safe, therefore the thickness of the pipe is set at 1/8 inch.

BUCKLING

$$P_{cr} = \frac{2.046\pi^2 EI}{L^2} \quad (3)$$

I is now known to be 6.28. E for A1 6061-T6 is 10,000 ksi. L is, worst case, 5.41 meters or 17.8 ft. P is given as 340 lbs. Solve to see if L checks out. Plugging in the values above yields a critical load of 4002 lbs on the moon. Thus there is a safety factor of 12 against buckling.

CABLE: worst case cable analysis suggests cable thickness of less than 1/16". Again to satisfy, uncertainties in these calculations a thickness of 1/16" will be used.

3.1 OVERVIEW

As part of man's continuing effort to explore and expand the frontiers of space, it is believed that one day man will establish a permanent outpost on the lunar surface. Once established on the moon, significant long term research can be undertaken in an effort to enhance current understanding of the lunar environment and the environment of space. To achieve this level of activity, it is necessary to develop some specialized tools and equipment suitable for operation in the lunar environment. Among the desirable pieces of equipment is a drilling apparatus capable of boring into the lunar surface for the purpose of collecting soil samples, inserting transducer probes, or locating ore deposits.

3.2 INTRODUCTION

3.2.1 Problem Statement

Our task was to create a conceptual design for a drilling apparatus to be used on the lunar surface. The rig will most likely be operated in the depression of a crater and under the cover of a large structure (geodesic dome). The desired drilling operations will not be massive excavations, but rather will be small bores for regolith samples, settings for structural anchors, instrumentation insertion into the lunar surface, or other similar purposes. Currently, there is very little information available regarding drilling operations on the lunar surface, and the limited information that is published is applicable only for a specific Apollo landing site. Therefore, many of the functional requirements and system specifications for this design are open-ended. Specific requirements will undoubtedly

become more clearly defined as more research is performed in this area. The general driving parameters for our design are listed below:

- functional in low gravity environment
- fit within structure measuring 30 meters diameter, 30 meters high
- total system weight not to exceed 6 metric earth tons (1 metric moon ton)
- modularized design for ease of transportation and assembly
- tele-operated with no direct human intervention
- drilling depths up to 100 meters
- drilling orientation vertical up to $\pm 15^\circ$
- power consumption less than 25 kW

3.2.2 Design Summary

The basic design of the drilling rig can be broken into four major components, with each component containing many sub-components. The four major components are:

- drilling rig platform
- turret
- robot arm
- drill string

Figure 1 shows a completely assembled sketch. Each of these separate components can be transported and handled easily as a single entity. With limited effort at the work site, the drilling rig components can be assembled together to form the working system.

The platform consists of a triangular truss structure with three leveling legs (section 3.3.1). The platform provides the "foundation" on which the remainder of the

drilling apparatus is built. The turret is positioned on top of the platform, and provides rotary motion for the robot arm (section 3.3.2). Through the center of the turret and platform is a drilling hole, which provides access to the lunar surface directly under the drilling rig. This area beneath the rig is the point of entry for all drilling operations. A hinged footplate within the turret drilling hole holds the drill string stationary while new sections of drill string are added or removed (section 3.3.5). The robot arm is attached to the rotary turret and is responsible for drill string manipulation (section 3.3.3). The arm consists of three links, with a revolute joint between each link to provide a planar range of motion.

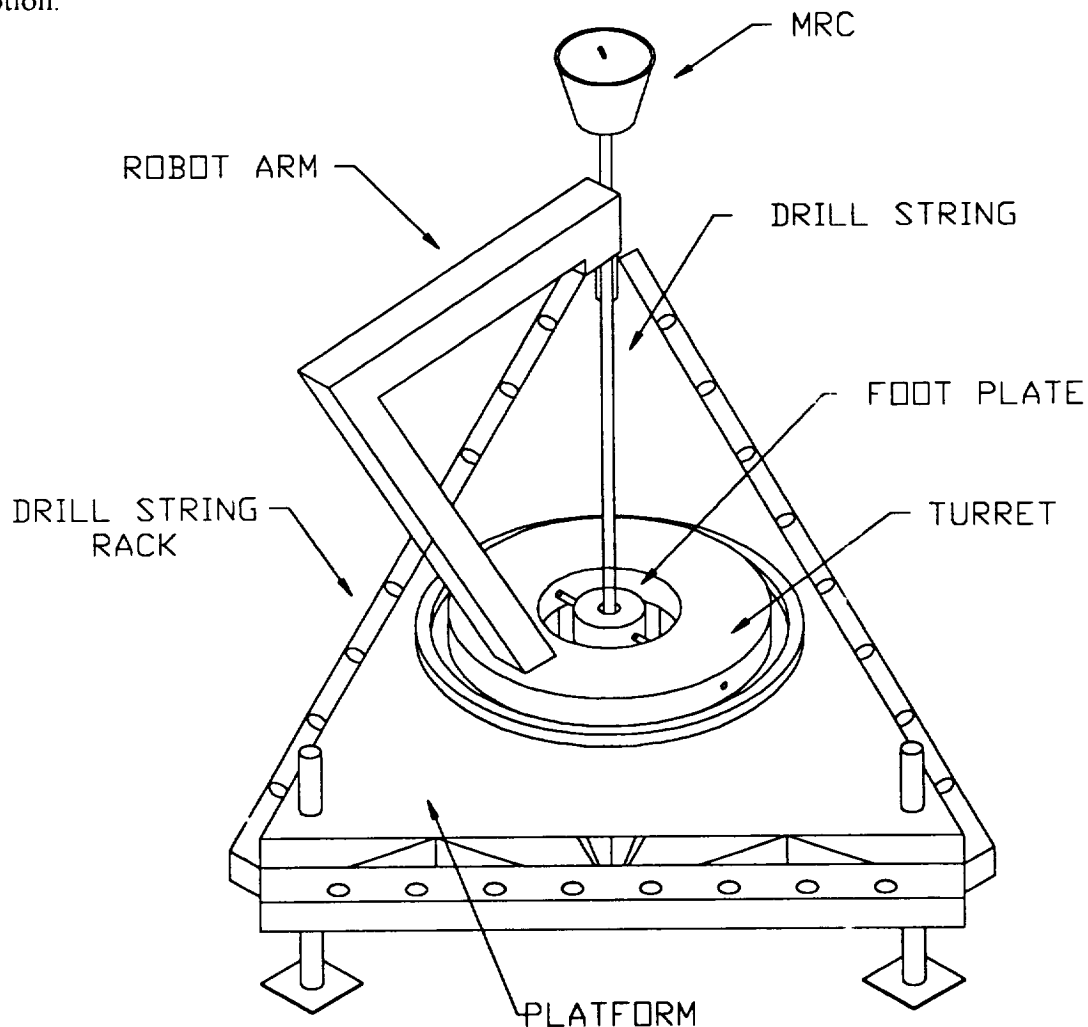


figure # 3-1

As mentioned previously, the turret provides rotary motion for the arm, thus giving it a full range of motion for retrieving or replacing drill string sections. The end effector on the robot arm provides the rotational power for the drill string, while the robot arm and the drill string weight provide the downward force on the drill bit during drilling. Furthermore, the robot arm end effector provides the power necessary to rotate the turret and level the platform legs, thus eliminating extra motors and reducing the power requirements for the entire system. Drilling operations can be performed in a vertical orientation up to $\pm 15^\circ$. The robot arm can position the drill string at various angles, and the hinged footplate can be locked in position to hold the drill string when drilling operations stop to add or remove drill sections. The drill string is designed in 5 meter long sections and each section is threaded at the ends so that multiple sections may be coupled together to reach 100 meter depths. The drill bit itself is located at the end of a drill string section (section 3.3.4). It consists of multiple cutting wheels arranged on a hollow cone, with the wheels at the bottom (tip) of the cone oriented vertically. The cutting wheels are arranged up the side of the cone, becoming angled "outward", where the cutting wheels at the top of the cone are oriented horizontally. Cuttings from the drill bit are carried up to the surface via a unique stepped auger within the drill string casing. A material removal container which is attached to the top of the drill string collects the cuttings and is periodically removed and emptied.

An aluminum alloy (6061-T6) was chosen as the material for all components with the exception of the drill string. Previous research shows that aluminum alloys have been successfully used on previous space missions. Its low density and high strength make it attractive as a structural material, and cost is relatively inexpensive. Other materials investigated were titanium alloys, aluminum-lithium alloys, and metal matrix composites (MMC). The other two alloys display undesirable properties. All bolts, pins, gears, etc. are made from tempered steel. The drill string is also made from steel to endure the loading from the drilling process and provide added downward force on the drill bit.

The physical size of the drilling apparatus fits well within the housing structure, and the total weight is 25% under the maximum weight limit (see Table 1 in Appendix B). All power transmission is achieved through electrical motors and geared transmissions, with no internal combustion engines and no hydraulics. Operation of the rig can be controlled by remote tele-operation with the use of a camera vision system or by a completely autonomous control algorithm programmed into the system.

3.3 COMPONENT DESIGNS

3.3.1 Platform

The overall purpose of the platform is to provide a stable base on which to support the actual drilling machinery. The drilling machinery consists of the following components: turret, on which the robot arm rotates, bearings, on which the Turret rotates, robot arm, and sections of drill string.

The design is free-standing and possesses the ability to be leveled. The power for leveling the platform is provided by the motors incorporated in the robot arm and is 100% automated. Also, the platform is designed with incorporated racks to hold the drill strings, which are not in use, and to provide the robot arm easy access to each drill string.

The overall concept of the platform has an equilateral triangle shape with each side of the triangle measuring 6 meters long. There is a one meter diameter hole in the geometric center of the platform. This will be defined as the drilling hole. The thickness of the platform is 1/2 meter thick. The structure consists of two aluminum plates, each 9.53 millimeters thick, with truss structures between the two plates. The first three trusses support the top and bottom plates by running along each edge of the plates. Each truss member has a 38.1 mm square cross section. There is another triangular truss system connecting the midpoints of each side of the plates. This gives a total of six regular truss

systems. The final support consists of a circular truss around the drilling hole. The drawings for the final design of the platform can be seen in **Appendix A figures 3-2 and 3-3**.

The supports for the platform consist of three legs. Each leg is positioned one meter from the corner of the triangle. The legs are fabricated from aluminum and are threaded on the outside with power screws. The top of each leg is grooved so that the end effector is able to interface with each leg. Their dimensions are 2 meters in length and have an outer diameter of .152 meters. The ends of the legs are interfaced with flat aluminum plates which in turn rest on the ground. The connection between the legs and the plates is a ball and socket joint.

The last component of the platform consists of a drill string rack. The drawings for the rack can also be seen in **Appendix A figures 3-2 and 3-3**. The rack is attached to the platform along each side. The rack holes are angled at 20° from vertical.

The platform is supported by three legs to make the base inherently stable. By choosing the three-leg design, a triangular platform is used to maximize area while providing a symmetry to distribute forces evenly. The length of the platform for the equilateral triangle (6 meters) accommodates all the machinery and provides a large amount of workable area. The top and bottom cover plates on the platform increase the rigidity while also providing a surface to carry additional equipment. The thickness of the platform (1/2 meter) allows the platform to be rigid and yet not too thick to allow for the free motion of the drill string. The use of the trusses increases the strength to weight ratio of the platform. Since the weight of the machinery is concentrated along the drilling hole opening, adding the circular truss distributes forces at this point.

The power screws on the legs allow the platform to be leveled automatically. This is accomplished by the robot arm. The maximum power required to operate the power screws is 1.1 kW (See **Appendix B**). As stated, the robot arm interfaces with each leg. By spinning the leg clockwise the robot arm raises the level of the platform and by

spinning the leg counterclockwise the robot arm lowers the platform. The ball and socket joint allows the leg to remain vertical even though the terrain might not be level.

The drill string rack is incorporated onto the platform to minimize the number of separate assembly components. The robot arm is able to reach each corner of the platform and therefore able to retrieve each drill string. The reasoning behind the angling of the rack is to keep the drill string from interfering with the rotational motion of the robot arm. Justification of the 20° angle is shown in **Appendix A figure**

3.3.2 Turret

The turret serves as the interface between the robot arm and the drilling rig platform. The main function of the turret is to provide rotary motion of the arm around the drilling hole of the platform. This allows for multiple drilling operations at various orientations without moving the rig platform. The turret structure also incorporates a footplate. The footplate is a clamping device through which the drill string passes. The purpose of the footplate is to clamp and hold drill string sections while the robot arm is attaching or removing a section of drill string.

The turret is a flat cylinder with an outer diameter of 2.25 meters (See **Appendix A figure 3-4**). The drilling hole, a 1.00 meter diameter hole through the middle of the turret, allows the drill string to pass through the turret center, through the platform (which also has a 1.00 meter diameter hole), and into the lunar surface. Located at the outer circumference of the turret is a large, reinforced collar for the robot arm connection. The footplate is a flat cylindrical component with a 0.5 meter outer diameter. There is a 150 millimeter hole in the center of the footplate through which the drill string sections are passed. Within this hole are three retractable chucks spaced 120° apart, capable of clamping and holding the drill string. Both the turret and footplate are 0.25 meter thick. The footplate is positioned within the turret drill hole and is attached to the turret via a

hinge passing radially through the footplate into the sides of the turret drill hole. The turret is mounted on the center of the platform directly over the drilling hole. A circular geared rack is fixed to the upper surface of the rig platform around the circumference of the turret. The end effector interfaces with the circular rack to provide rotational motion of the turret. Commercially available flat roller bearings are used between the turret and platform.

By interfacing the footplate directly with the turret (and therefore to the robot arm), the footplate rotates with the turret structure. The simple planar motion of the robot arm relative to the turret only requires a simple one degree of freedom hinge on the footplate. This hinge allows the footplate to be angled and locked in positions up to $\pm 15^\circ$ to accommodate non-vertical drilling operations. The robot arm end-effector is able to interface with the circular rack on the platform with a rack and pinion type interface, and rotate the entire turret structure and arm about the drilling hole. This approach requires no additional power sources and no additional moving parts. The retractable sections within the footplate clamp and hold the drill string sections when drilling operations stop. This occurs each time a new section of drill string is being attached or removed. As a drill string section is being attached or removed, the footplate grasps the lower drill string to prevent rotation while the top section is being screwed on or off. Furthermore, when a long drill string is removed from a hole, the footplate will hold the string as sections are removed, preventing it from dropping down into the hole.

3.3.3 Robot arm

A three link robot arm is located on the turret. The main purpose of the arm is to drill and manipulate drill string sections. Since it is desirable to have only one motor driven component, the arm is also responsible for several other operations. The arm

provides power to also level the platform, rotate the turret, and retrieve/replace each drill string during the drilling operation.

The robot arm has three links with a motor at each joint for planar mobility. On the third link the arm has an end effector as shown below in **figure 3-5**.

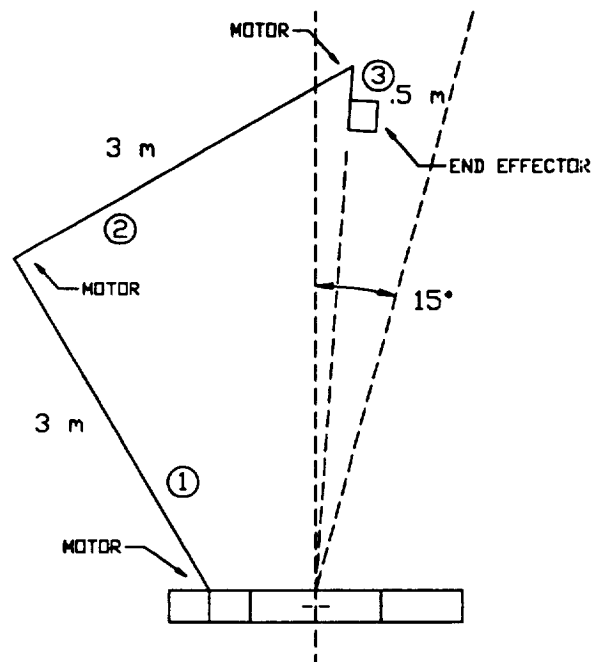


figure # 3-5

Links (1) and (2) are both 3 meters long and link (3) is 0.5 meters long. Each link is made of square tubing. The square tubing is .305 meters on a side and 12.7 mm thick. The arm is made out of aluminum (6061-T6).

As stated, the main functions of the robot arm are to drill and manipulate drill string sections. In order to drill the arm must be able to reach drill string sections from the rack and position them vertically over the drill hole. The arm is able to reach any point on the platform as shown in **Appendix B**. It must also be able to grab and release the drill string. This is accomplished with the help of the end effector on the third link of the arm. The end effector is a hollow cylinder that works like a power drill. The cylinder is rotated using a motor and there is an inner ring which can be stopped allowing four sleeves to

tighten or loosen depending on the direction the cylinder is rotated. This can be seen in **Appendix A figure 3-6**. The minimum power required to rotate the entire drill string is calculated to be 5.35 kW, which is the sum of the power required to rotate the drill bit and drill string.

The robot arm is able to level the platform using the three adjustable legs. Each leg is a power screw and the top of the leg is geared the same as the bottom of the end effector. The arm is able to reach each leg in order to level the platform. The description can be seen in **Appendix B**.

The robot arm is also able to rotate the turret on which the arm is mounted. There is a geared rack mounted onto the platform outside the turret. This rack is geared the same as the bottom of the end effector. When the arm and turret need to rotate, the arm places the end effector against the geared rack. The end effector is then rotated causing the turret and arm to rotate to a specified position.

The arm is sized through the use of a dynamic analysis on each link to determine both the normal and shear stress. The analysis can be seen in table 2 of **Appendix B**. A spread sheet is used to test several different drilling positions and to test how fast the arm could rotate and accelerate. The critical operation is the drilling. During the drilling the force exerted downward is 1/2 the weight of the platform. This causes the largest shear and normal stresses. Therefore, drilling at several different angles are studied to determine the correct thickness of each link. The correct thickness is found by comparing the calculated stresses to the yield stress of aluminum 6061-T6 which is 270 MPa. The thickness is therefore found to be 12.7 millimeters for a .305 X .305 meter square tube, which results in a maximum stress of 229 MPa. Several of these tests are shown in **Appendix B**.

The initial task of the robot arm is to level the platform. Next, the arm must retrieve a drill string from the drill rack and begin drilling to a specified depth and angle. The arm can drill up to one hundred meters and has a range of angles of plus or minus

fifteen degrees from vertical. The calculations for the below ground work envelope are shown in **Appendix B**. Upon completion of the drilling operation, the arm pulls the drill string up section by section and returns each section to the drill rack.

3.3.4 Drill Bit

To bore through the surface of the moon, some type of drill bit is required. This drill bit must be able to bore through the lunar surface with a bore hole diameter of 100 millimeters. The design that is discussed will accomplish this task and also provide a means in which to begin the material removal. During the early stages of the drill bit design, it was decided that the drill bit design would pursue a concept that would incorporate multiple cutting wheels. These wheels, when properly oriented, will cut small grooves into hard solid surfaces and cause the solid to "chip off" in narrow regions between grooves. These cutting wheels operate much like a glass cutter, concentrating a large force over a small region (the cutting edges) to produce a high stress. Furthermore, a similar approach is successfully used in terrestrial mining on large boring devices used to drill through granite rock, and other various solid substances. The final drawing for the bit are shown in **Appendix A figure 3-7**.

The overall shape of the bit is a hollow cone with a base diameter of 100 millimeters. The angle of the cone will be 90° which is used to determine the height. This height is 50 millimeters. A helical groove of material will be removed from the bit on the outside. The width of this groove is 2mm, the height is 5mm, and it ascends the cone at an angle of 10° . The cutting wheels are contained within the groove. The cutters are oriented with an increasing angle whereas the last set of cutters are angled 45° relative to the trace of the drill string. The initial cut will be performed by four cutting wheels (diameter 6.35mm) on the tip of the bit. There are 21 identical cutting wheels equally spaced throughout the groove. Also, within the removed helix, there are openings that are

cut perpendicular to the face of the cone which are equally spaced behind each wheel.

An internal auger is contained inside the bit. This auger is responsible for beginning the material removal process. It is aligned with the drill string and matches the angle of the drill string's auger. The bit is also threaded for easy coupling and de-coupling with any drill string.

Because much of the deeper lunar soil is hard packed and rock-like, it seemed logical to approach the drill bit design in a manner that would incorporate the glass-cutter design. An additional benefit also be inherent in this cutting wheel design is the lower torque requirement needed to rotate a drill bit with rollers, as opposed to a spiraled auger drill bit. The values for torque and power are 110 Newton-meters and 3.46 kW respectively. Calculations for torque and power requirements are shown in **Appendix B**. This is important due to the limitations of power supply. The openings that are cut in the removed helix groove allow the cut material to fall into the center of the bit.

Once the bit is attached to the drill string, there is no need for any interaction with any other component. The bit will bore at a rate of 78.6mm/s (See **Appendix B**). The cut material should proceed into the center of the bit and be transported up the auger by a combination of rotational and longitudinal motion.

3.3.5 Drill String

The drill string is one of the most critical sub-systems of the entire drilling rig, and consequently, it is one of the most intricate components. The purpose of the drill string is two fold. First, the drill string sections provide the structural members necessary to rotate the drill bit and propel it down into the lunar surface to a depth of 100 meters. Secondly, the drill string must provide a means to transport debris from the site of the drill bit to the surface during the drilling operation.

The drill strings are designed in 5 meter long sections, threaded at each end for secure coupling and simple disconnection between multiple drill string sections. Each drill string section consists of a cylindrical outer sleeve encasing an inner solid shaft with an auger (See **Appendix A figure 3-8**). The entire component is rigid and rotates as one piece, that is, there is no relative rotation between the outer sleeve and the inner shaft/auger. The cylindrical sleeve measures 95 millimeter outer diameter with a 5 millimeter wall thickness. The inner shaft is solid with a 15 millimeter diameter. On this shaft, there is a 30° auger-like spiral which is also attached to the inner wall of the outer sleeve. On the upper surface of the auger, there are "gates" - 20 millimeter high vertical walls welded to the auger in a radial orientation from the inner shaft to the outer sleeve. These gates will be spaced 60° apart, over the entire height of the spiral auger. Connection of the drill string segments ensures a proper interface between the spiral auger sections, to form one continuous auger. The entire drill string is fabricated from a structural steel.

The 5 meter length is selected because it is easily manageable with a moderately sized robot arm, convenient for portability, and capable of reaching 100 meter depths with 20 to 25 coupled sections. Some extra drill sections are necessary to reach 100 meter depths when drilling in non-vertical orientations. Also, some drill string sections may be specialized with unique transducers affixed to the end, or replacement drill bits. The outer diameter of the sleeve is slightly undersized compared to the drill bit to reduce friction between the lunar bore and the rotating drill string. Calculations are performed to find power and torque requirements for drill string friction. These values are found to be 1.89kW and 60.2 N-m respectively and can be found in **Appendix B**. The drill string is fabricated from steel to sustain the possible high load and torque values produced during drilling operations. Furthermore, the added weight of the steel provides increased downward force on the drill bit which is desirable during drilling. Stress- strain failure analysis for three cases can be found in **Appendix B**.

During drilling operations, the drill string is periodically shaken in the axial direction. This may be accomplished by the robot arm, a cam follower device, or a spring loaded mechanism. This shaking procedure causes debris fragments from the drill bit site to "climb up" the auger. Debris is accelerated axially upward from the auger surface, impacting the adjacent auger layer just above and ricocheting (deflecting) back downward at an angle to the original auger layer at a slightly higher position. The gates on the upper surface of the auger serve as "steps" for the debris fragments to climb. As debris climbs the drill string, it exits at the top opening, into a cup-like device to be removed and emptied before the connection of each new drill string segment.

3.3.6 Material Removal Container

The material removal container (MRC) is a simple device to aid in the collection and removal of cuttings produced during the drilling process. This is important so to keep the drilling apparatus and drill site free from debris. It provides an effective method to neatly collect the cuttings in small, intermittent loads. The materials handling team can easily and quickly remove and empty the container, and the drilling procedure will continue.

The MRC is cylindrical in shape with a 15° outward sloping wall (See **Appendix A figures 3-9 and 3-10**). The diameter of the top of the MRC measures 0.7 meter, while the diameter of the bottom of the container measures 0.415 meter. The total height of the MRC is 0.527 meter. The top of the container has a partial circular cover with a small opening around the outer circumference to allow cuttings to be dumped. The cover is fastened to the top of the MRC by four thin rods. On top of the cover, there is a 45° "jingle bell" interface connection to permit lifting and hoisting. A 230 millimeter deep by 150 millimeter wide trough is located along the outer circumference surrounding the 80 millimeter hole through which cuttings exit the drill string and enter the MRC. In the

center of the bottom of the MRC there is a concentric sleeve which slides into and over the end of a drill string. The cup clips onto the top end of the drill string through the use of four spring clips in the sleeve of the MRC spaced 90° apart. The entire container is fabricated from 3.175 millimeter thick aluminum alloy.

The 15° wall slope is used to facilitate the container emptying process. Since the materials handling team can only rotate the MRC 90°, the sloped side wall will provide increased angle for the cuttings to slide down. The concentric sleeve provides the most rigid and slop-free interface between the MRC and the drill string. This is important to maintain a secure connection during the axial shaking procedure. The useful volume (volume of the trough) of the container is approximately the same as the volume of 1 drill string section. The "jingle bell" connection is angled at 45° to permit the materials handling team to interface with the MRC at an angle, and to rotate it 90° to empty it. The amount of elapsed time between MRC emptyings is unpredictable due to the nature of the axial shaking mechanism and the movement of cutting chips up the stepped auger. Clearly, the MRC is removed and emptied at least before each new drill string section is added during the drilling process. Spring clips are used to attach the MRC to the drill string because they provide an adequate holding force while allowing for simple push on attachment and pull off removal. The material chosen for fabrication of the MRC is aluminum because it is lightweight, inexpensive, and rigid enough for this application.

3.4 OPERATION

The operational phase of the drill rig begins after the platform and all machinery is set in the position of operation (PO) by the materials handling team. The actual drilling operation begins with the selection of the angle to drill. Once this angle (α) is chosen, the robot arm selects a drill string with a pre-attached drill bit on the end. It should be noted

that $-15^\circ \leq \alpha \leq 15^\circ$ and is measured from the vertical. The robot arm end effector grasps the drill string. (See section 3.3.3) The next step is to attach the material removal container on top of the drill string. This is accomplished by a system of spring clips. (See section 3.3.6)

At this time the robot arm will spin the drill string at an angular velocity of 31.1 rad/sec, which corresponds to 300 rpm. This is coupled with a downward force of 4400 Newtons, one half the maximum force allowed the free standing structure. If needed, a higher force could be provided by anchoring the platform legs to the ground. The robot arm begins the angular motion at a position of four meters above the platform and drives the drill string down for one meter. This is defined as one *drilling stroke*. Once the robot arm has completed one stroke, the axial shaking occurs for a sufficient amount of time. Next, the footplate will grasp the drill string and hold it secure while the end effector releases the drill string. The robot arm then moves up the straight line trace a distance equal to the drilling stroke. The end effector then reattaches itself to the drill string. After the footplate releases the drill string, the drilling operation continues.

When the drill string is imbedded into the lunar surface to a level where the top of the string extends a distance less than 1 meter above the platform deck, a new section should be added. This is the cue for three simultaneous operations. The footplate grasps the drill string. Once this is accomplished, the end effector disengages the drill string in use and the robot arm retrieves another section of drill string from the drill string rack. This is defined as *drill string changeout*. At this time, the MRC is removed by the materials handling group and emptied. Periodically, it is necessary to remove and empty the MRC at times other than drill string changeout. As this is happening, the robot arm attaches the new section of drill string to the original section. Once the new section is attached, the material removal container is placed upon the new section. This process will proceed until the desired depth is reached.

At this stage of the design, control of the robot arm can be achieved in several ways. One method is by remote tele-operation incorporating a camera-vision system. Another is the use of a control algorithm which is pre programmed into the robot arm. Either system is feasible and a selection would be made after further study.

3.5 RECOMMENDATIONS

Due to imposed time constraints and limited manpower, all facets of our design project cannot be completely analyzed. The broad scope of this project and the large number of systems forced us to take a general approach to a high level design. All major components have been investigated, but clearly, more work should be performed. It is our recommendation that future groups use our report as a starting point, and continue the design. Perhaps, the project could be broken into several design projects and assigned to various design teams to investigate:

- overall control scheme
- materials research (aluminum, titanium etc.)
- bit wear
- labeling for depth of removed material
- more analysis on the legs for leveling (ball & socket joints, power screws)
- complete dynamic analysis of robot arm
- heat generation issues

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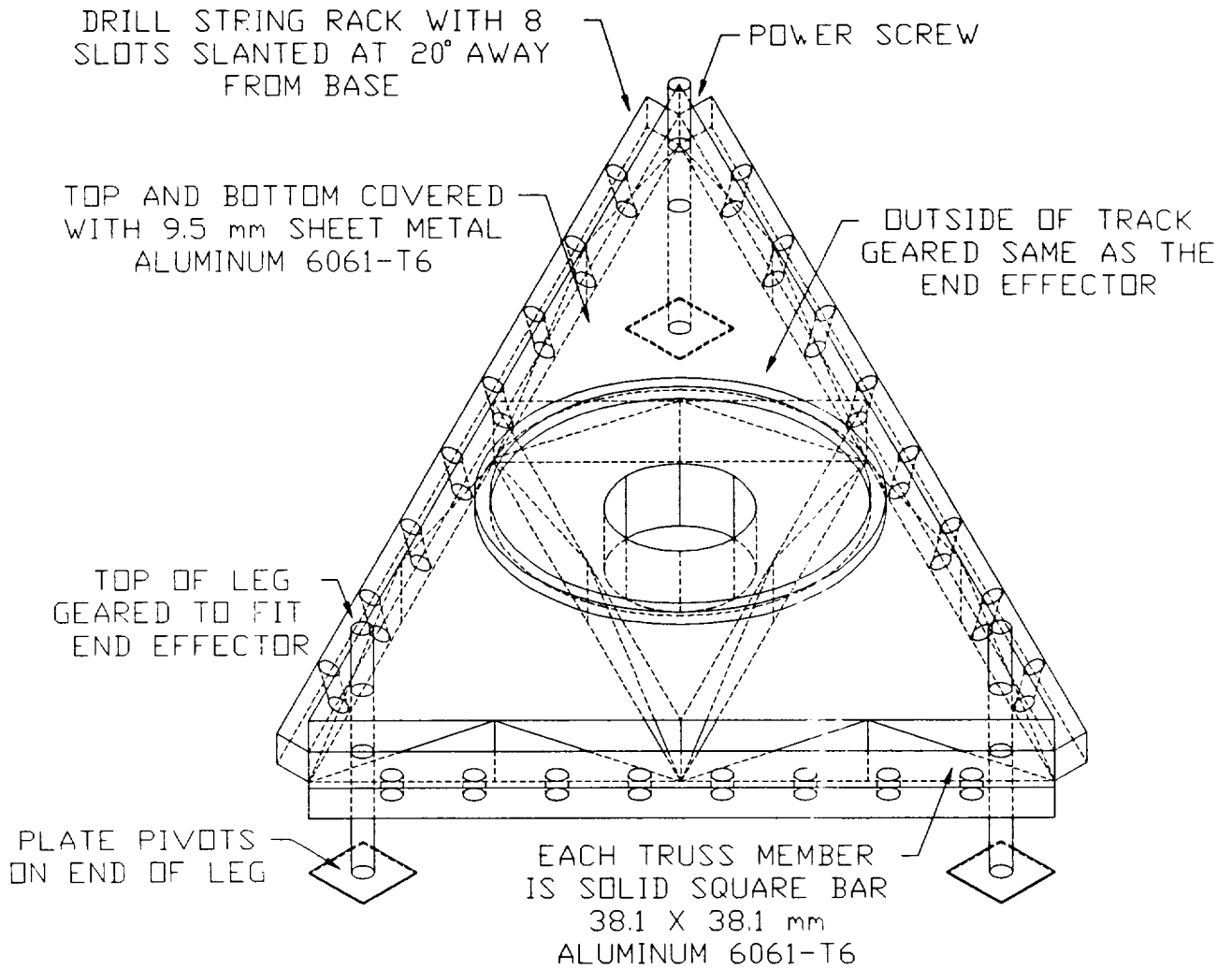
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3.7.1 APPENDIX A



LUNAR DRILL

DRAWING: 3.2

PLATFORM

DATE 3/10/94

SCALE: NONE

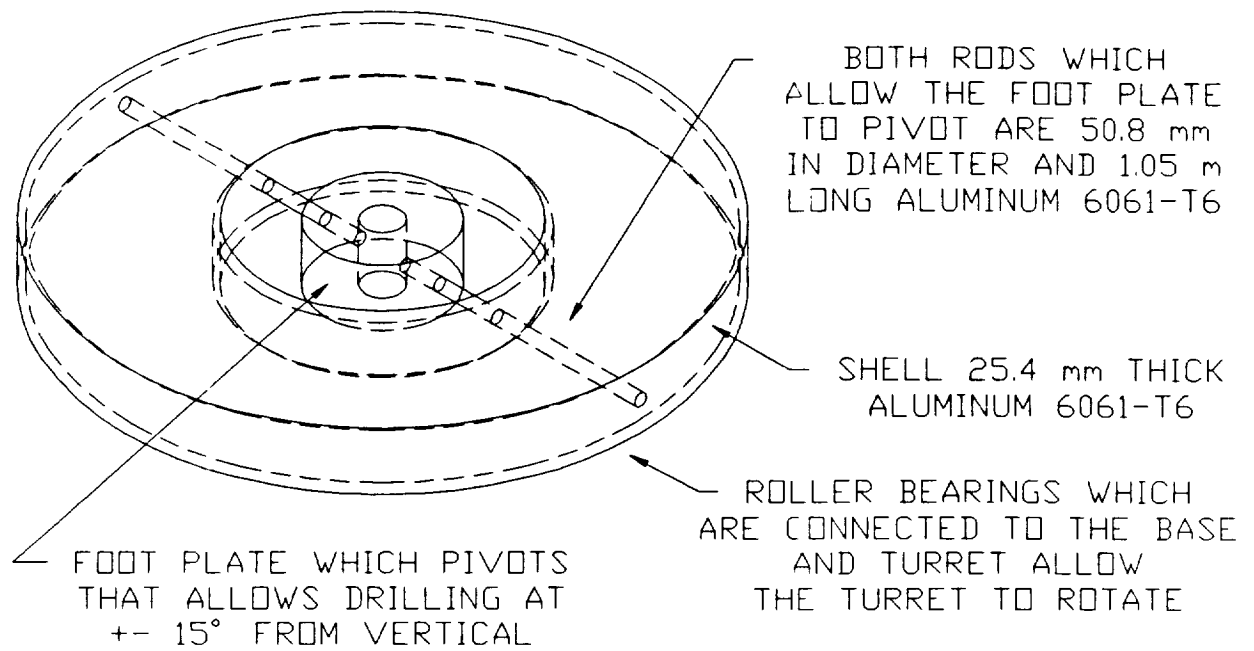
[illegible]

Diagram illustrating a structural truss system, specifically two equilateral triangular trusses joined at a central point. The truss is supported by a base with a 0.3 m gap on the right side. The truss members are 0.15 m thick. The central opening has a diameter of 1 m. The truss is labeled "TWO EQUILATERAL TRIANGULAR TRUSSES".

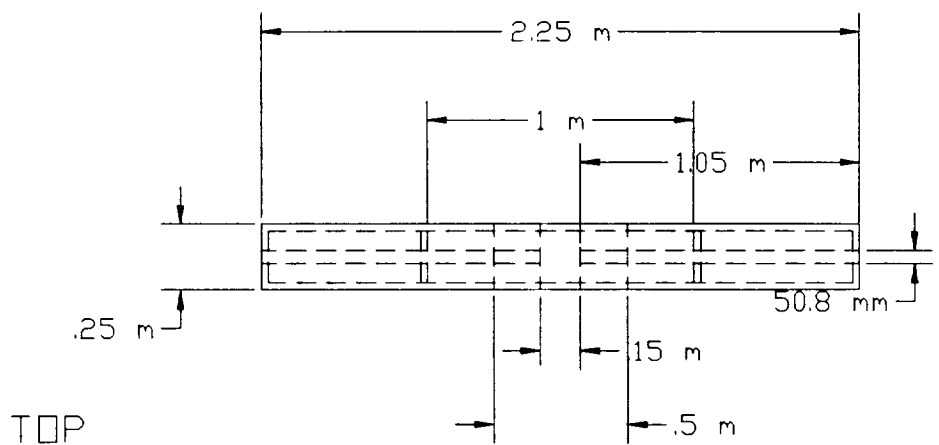
Key dimensions and features:

- Top chord width: 0.5 m
- Top chord height: 0.5 m
- Base width: 6 m
- Central opening diameter: 1 m
- Truss member thickness: 0.15 m
- Truss height (from base to top): 2.5 m
- Truss height (from base to top): 2.7 m
- Truss height (from base to top): 3 m
- Truss height (from base to top): 0.3 m

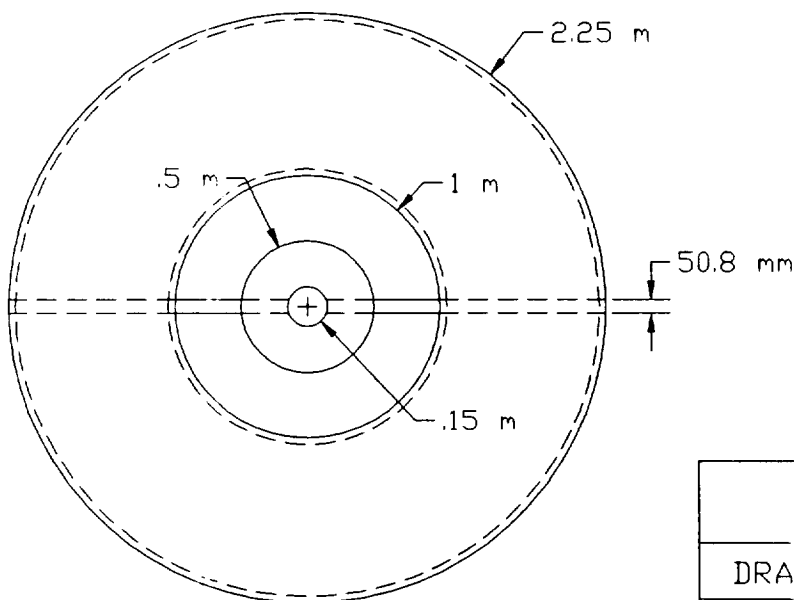
DRAWING: 3.3	PLATFORM
DATE: 3/10/94	SCALE: NONE



FRONT



TOP



LUNAR DRILL

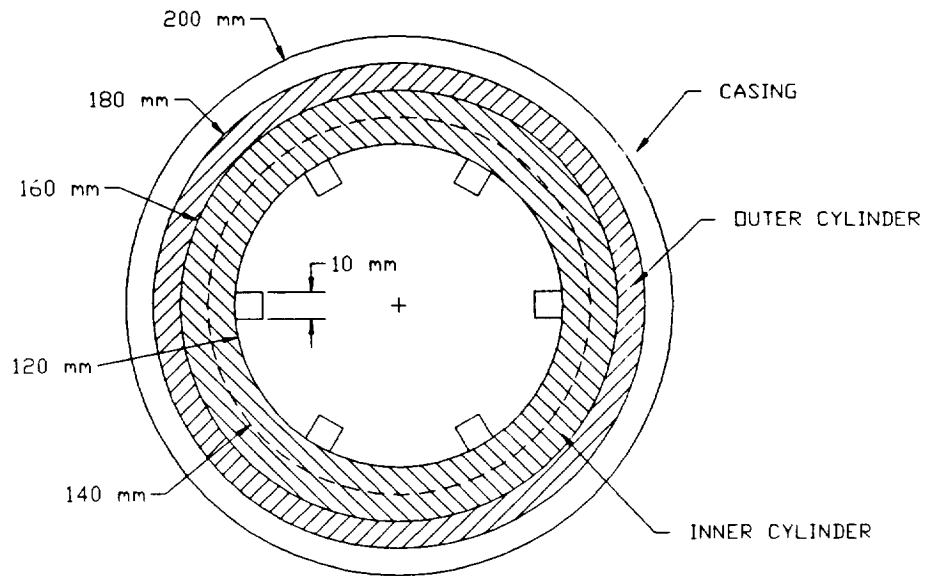
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TURRET

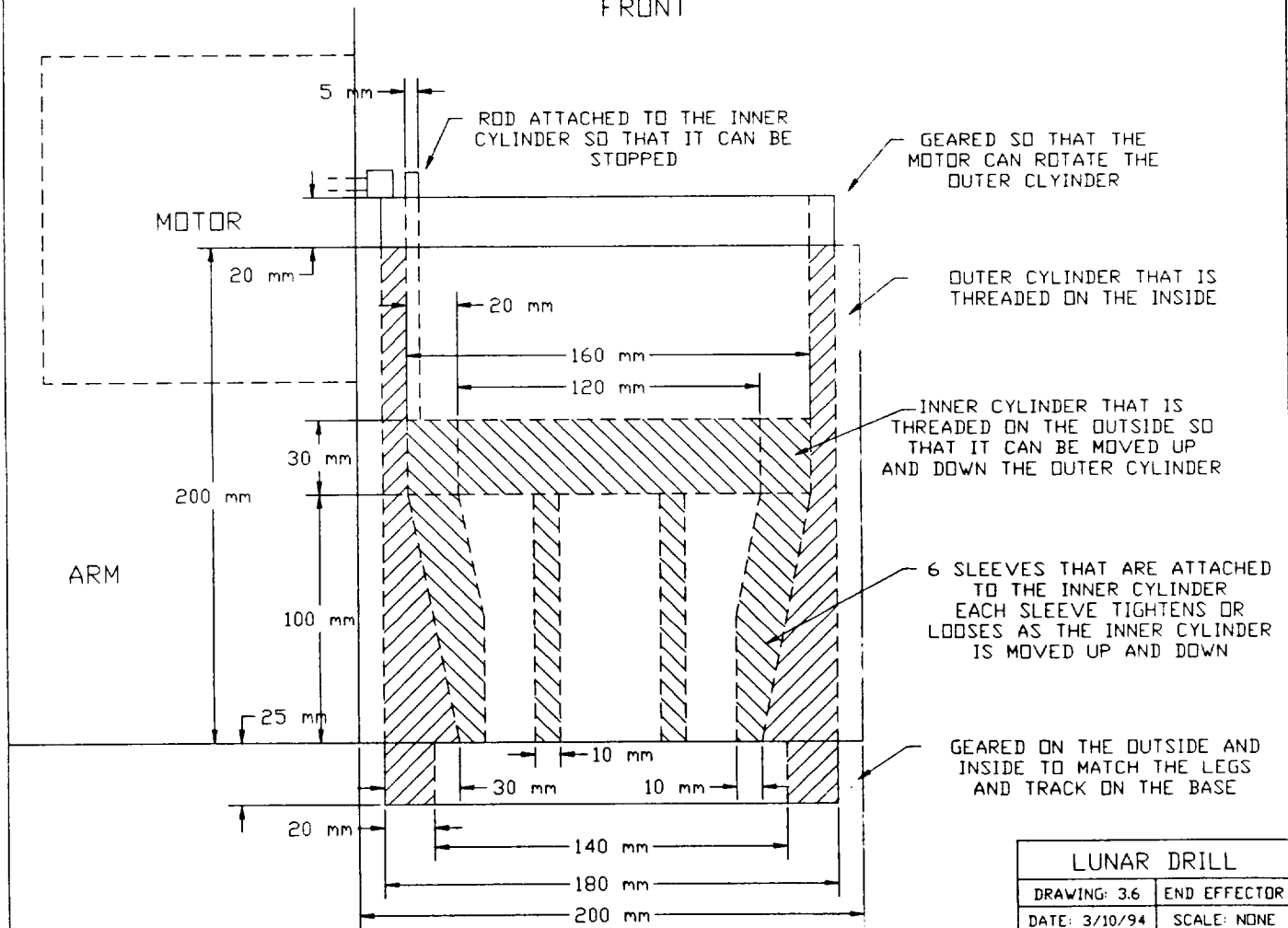
DATE: 3/10/94

SCALE: NONE

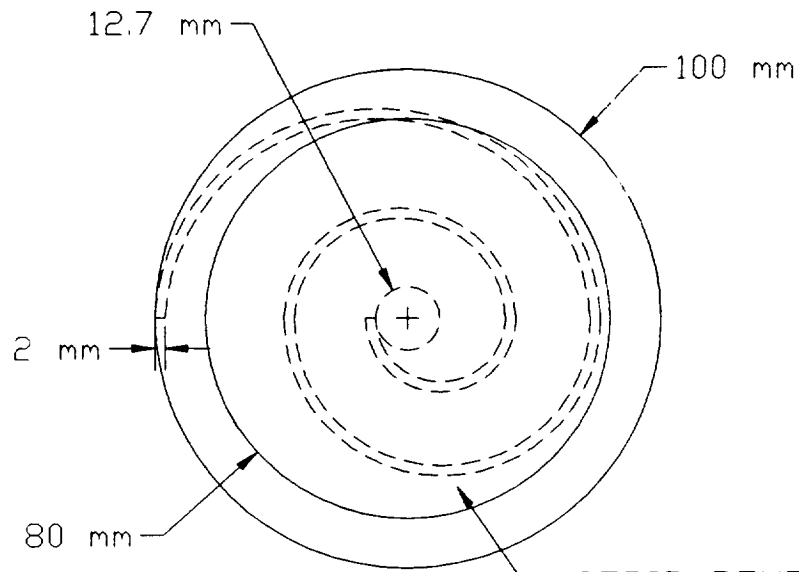
TOP



FRONT

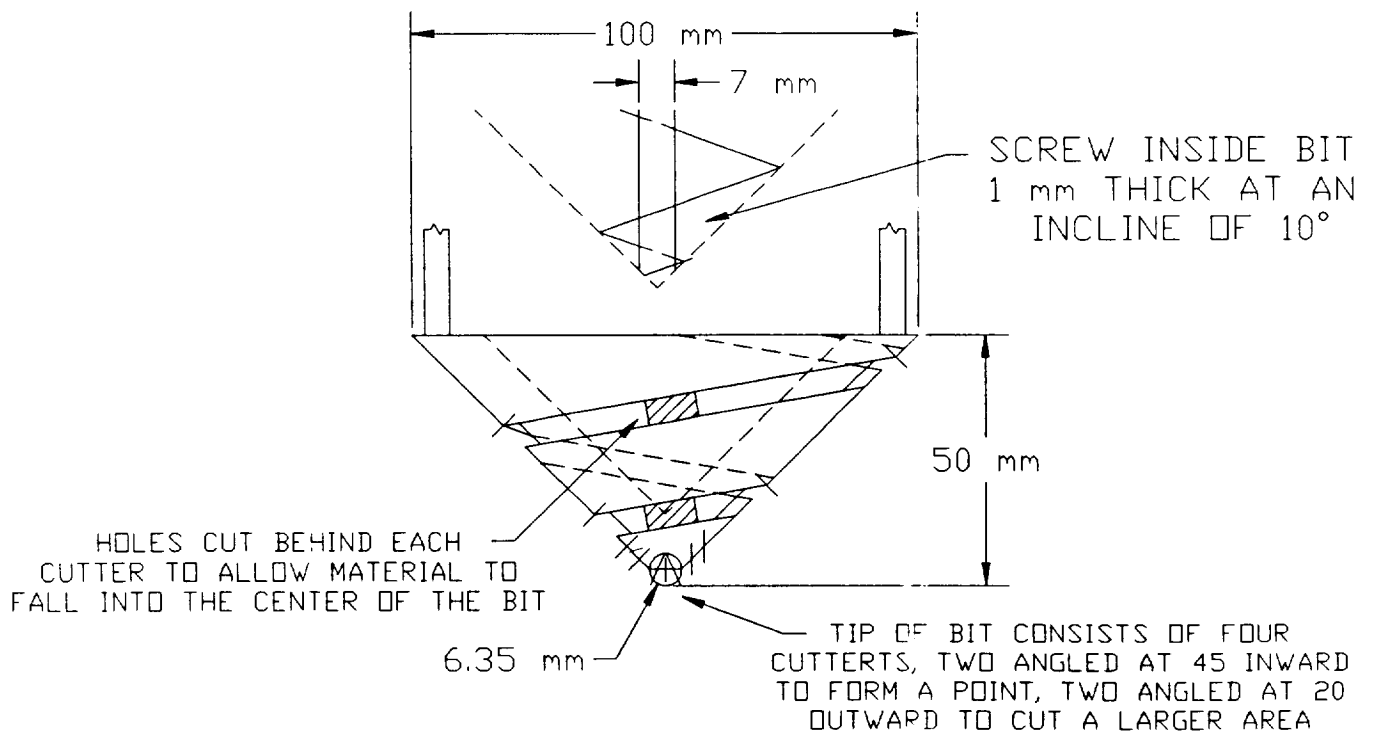


TOP



STRIP REMOVED FOR CUTTERS AT A DEPTH OF 2 mm A HEIGHT OF 5 mm AND INCLINED AT 10°

FRONT



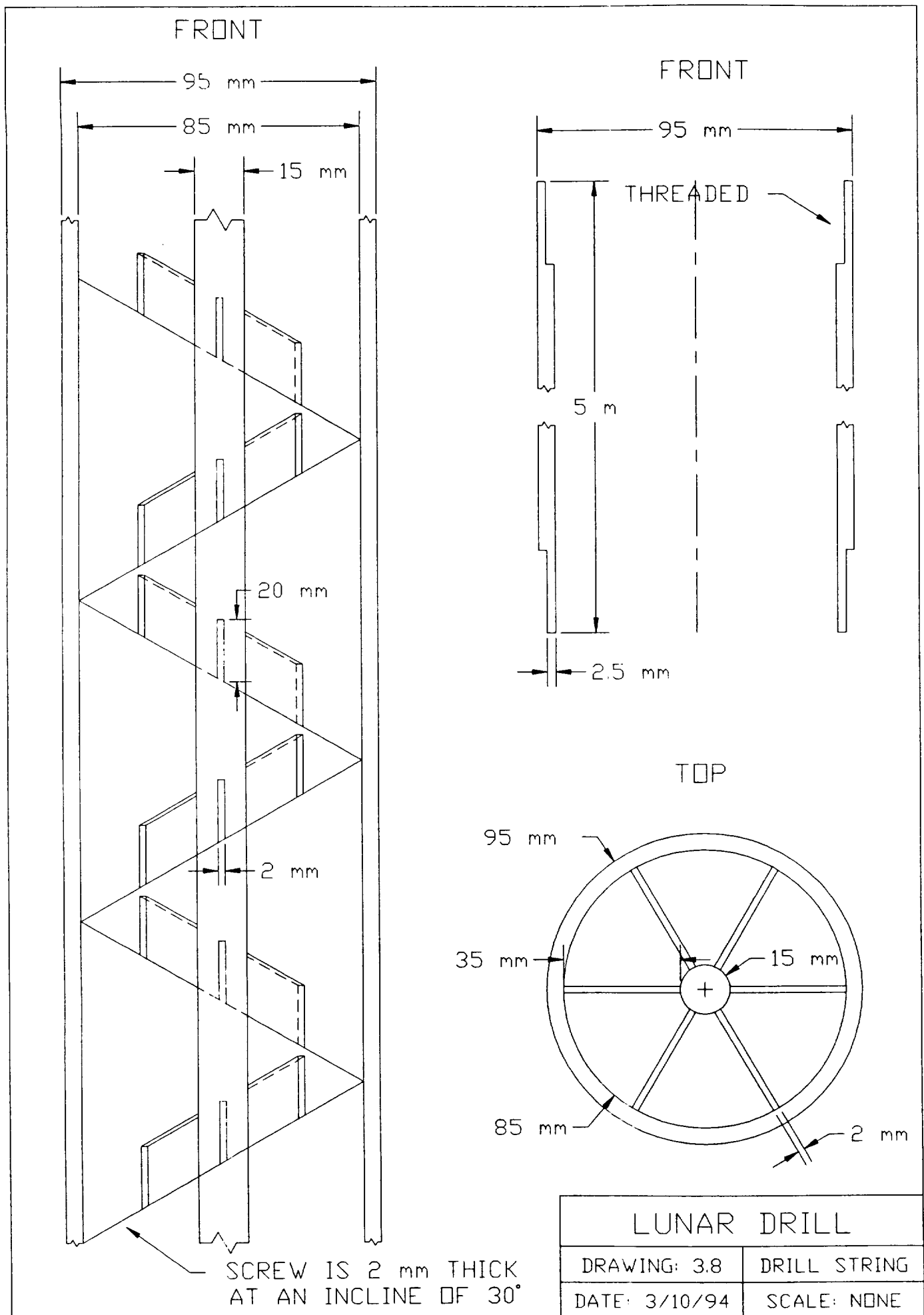
LUNAR DRILL

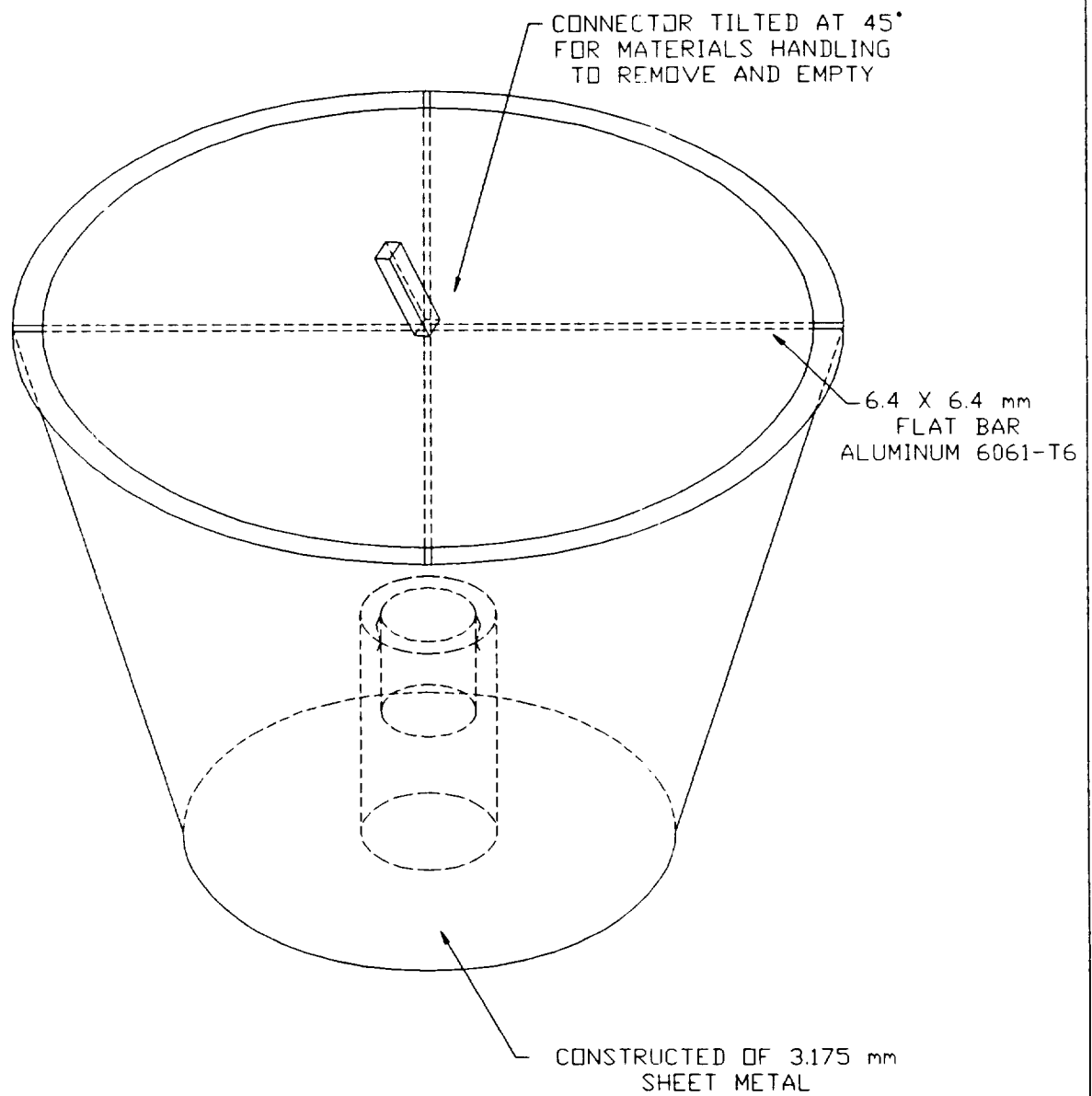
DRAWING: 3.7

DRILL BIT

DATE: 3/10/94

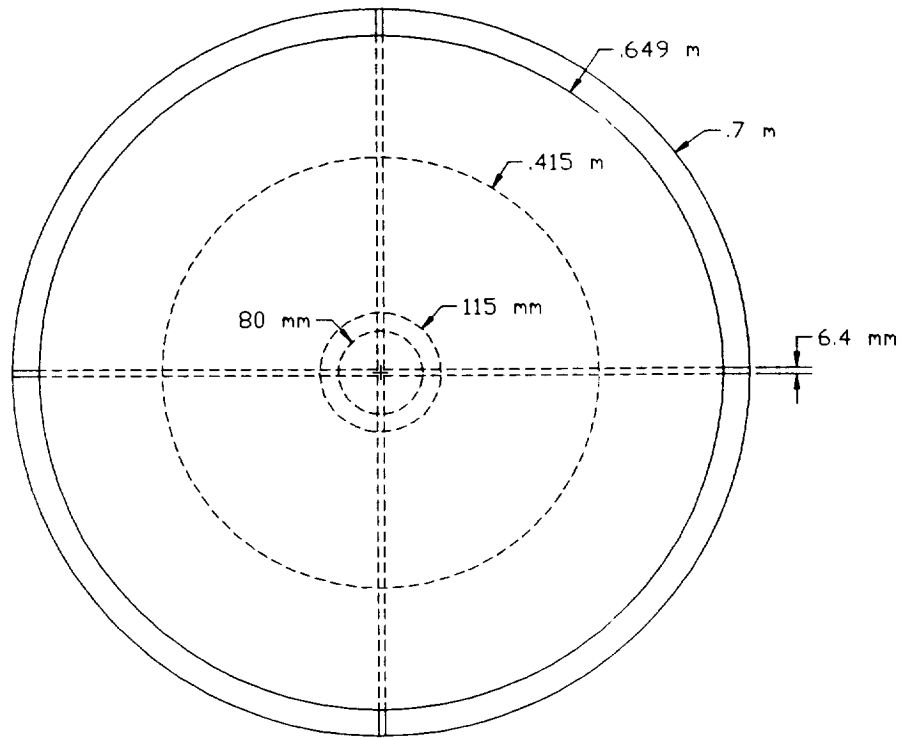
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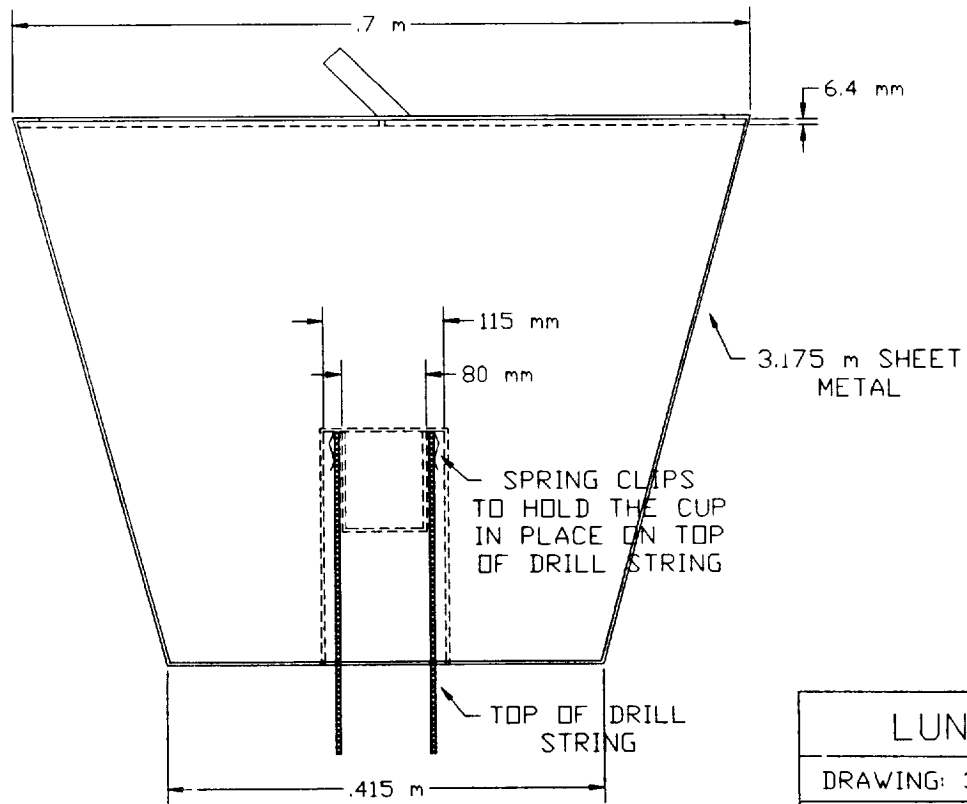


LUNAR DRILL	
DRAWING: 3.9	MRC
DATE: 3/10/94	SCALE: NONE

TOP



FRONT



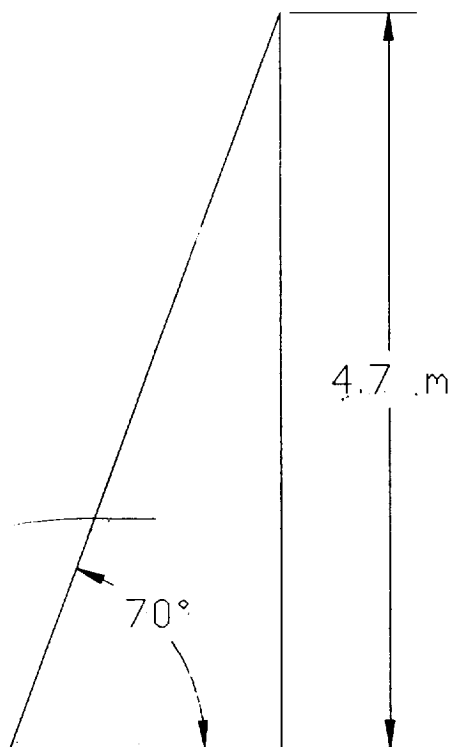
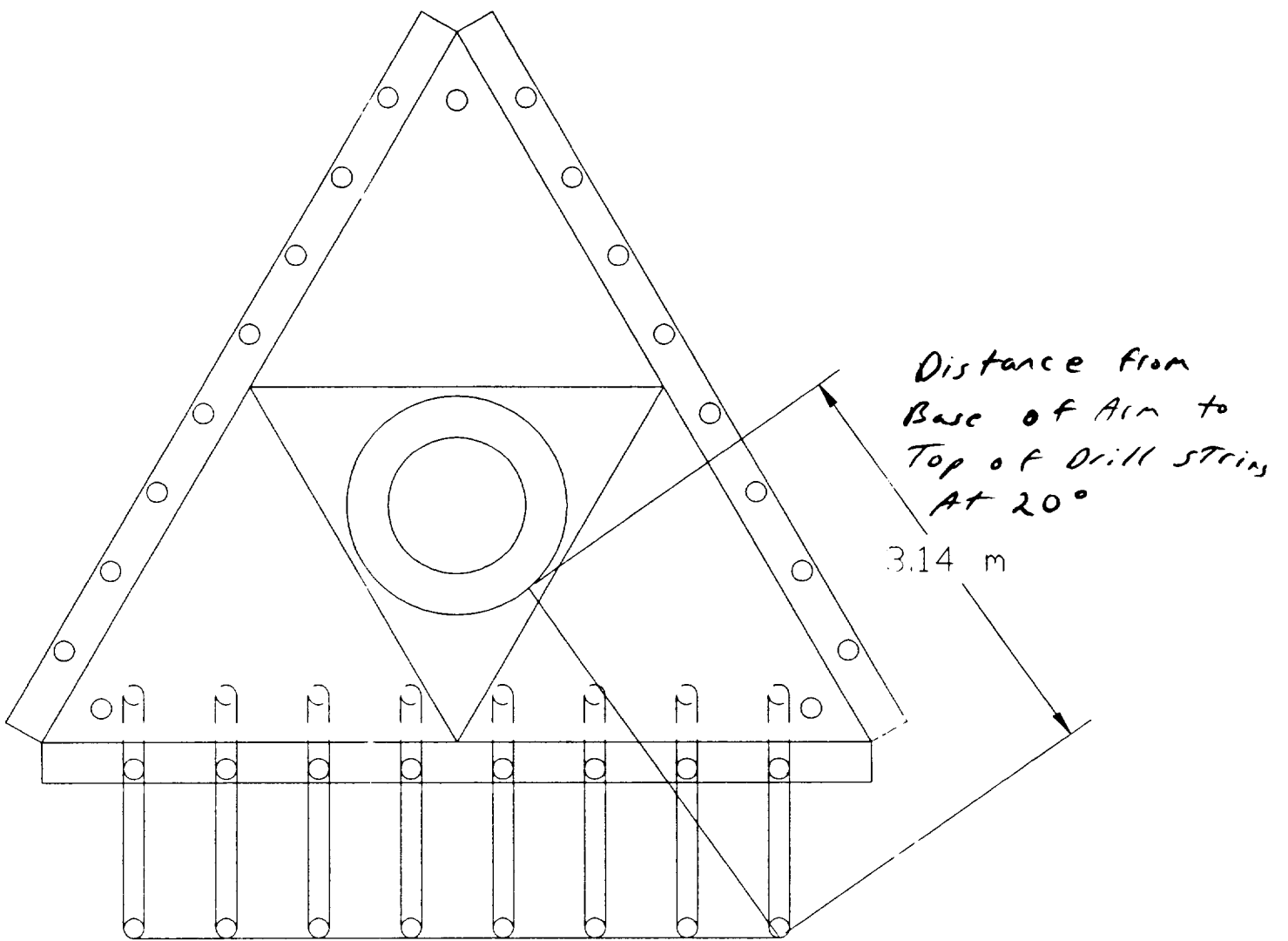
LUNAR DRILL

DRAWING: 3.10

MRC

DATE: 3/10/94

SCALE: NONE



Height of 1 Drill String at 20°

3.6.2 APPENDIX B (Calculations)

Square Thread Power Screw (Platform Legs)

Calculation for torque & power to raise and lower the platform

μ = coefficient of friction (power screw) = 0.15
 ℓ = lead = 0.0254 m
 d_m = mean power screw diameter = 0.1397 m
 F = load = 7280/3 = 2427 N
 T = required torque N-m

to raise the load: [Shigley, 1989]

$$T = \frac{F \cdot d_m}{2} \left(\frac{\ell + \pi \mu d_m}{\pi d_m - \mu \ell} \right)$$

$$T = \frac{(2427)(0.1397)}{2} \left(\frac{0.0254 + \pi(0.15)(0.1397)}{\pi(0.1397) - (0.15)(0.0254)} \right)$$

$$T = 35.55 \text{ N-m}$$

$$P = T \cdot \omega \quad \omega_{\max} = 5 \text{ rev/sec} = 300 \text{ rev/min} = 31.42 \text{ rad/sec}$$

$$P = (35.55)(31.42) = \underline{1.117 \text{ kW}}$$

to lower the load:

$$T = \frac{F \cdot d_m}{2} \left(\frac{\pi \mu d_m - \ell}{\pi d_m + \mu \ell} \right)$$

$$T = \frac{(2427)(0.1397)}{2} \left(\frac{\pi(0.15)(0.1397) - 0.0254}{\pi(0.1397) + (0.15)(0.0254)} \right)$$

$$T = 15.48 \text{ N-m}$$

$$P = (15.48)(31.42) = \underline{0.5 \text{ kW}}$$

Self-locking condition (load alone will not cause power screw to spin)

$$\pi \mu d_m > \ell \quad [\text{Shigley, 1989}]$$

$$(3.14)(0.15)(0.1397) = \underline{0.066 > 0.0254} \quad \text{so therefore power screw is self-locking}$$

Cutting Rate of Drill Bit

Cutting Rate = Depth of Cut * RPM

CR = DC * RPM

Upper Limit:

DC = 5 mm

RPM = 300 = 31.42 rad/sec

CR = (5 mm)(31.42 rad/sec) = 157.1 mm/sec

Lower Limit:

DC = 1 mm

RPM = 60 = 6.28 rad/sec

CR = 6.28 mm/sec

Expected Operating Conditions:

DC = 2.5 mm

RPM = 300 = 31.42 rad/sec

CR = 78.55 mm/sec

Drill Bit Torque & Power Requirements

- F = downward force per wheel
F_d = friction force on wheel
μ = coefficient of friction (soil against rolling wheel) = 1.0
r = radial position of wheel = 0.025 = r_{avg}
T = total torque from drill bit
P = power required from drill bit

$$F = \frac{\text{Total downward force}}{\text{\# of wheels}}$$

$$F_d = \mu F$$

$$T = \sum F_d \cdot r \quad \text{summed over all wheels}$$

$$P = T \cdot \omega \quad \omega_{\max} = 300 \text{ rpm} = 31.42 \text{ rad/sec}$$

$$F = \frac{4400 \text{ N}}{25} = 176 \text{ N per wheel (absolute maximum)}$$

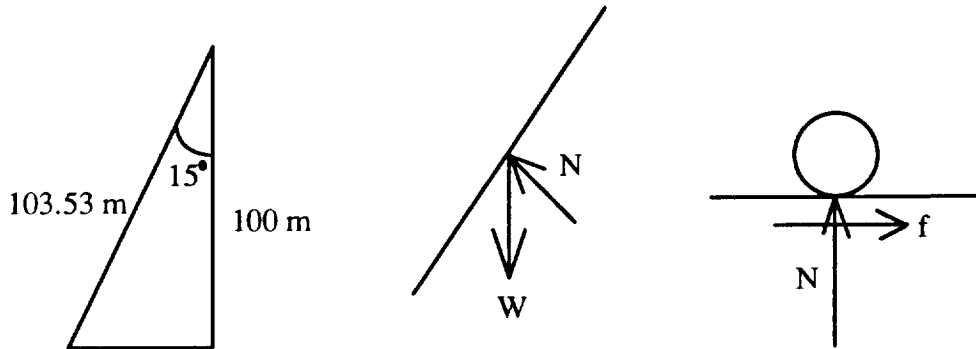
$$F_d = (1.0)(176 \text{ N}) = 176 \text{ N}$$

$$T = (25 \text{ wheels})(176 \text{ N})(0.025 \text{ m}) = \underline{110 \text{ N-m}}$$

$$P = (110 \text{ N-m})(31.42 \text{ rad/sec}) = \underline{3.46 \text{ kW}}$$

Drill String Friction Torque & Power Requirements

Worst case scenario: Drill string oriented at 15° angle with entire length of drill string contacting the bore wall.



21 drill strings to reach 103.53 meter length
mass of 21 drill strings = 1497 kg

m = mass of drill strings = 1497 kg
 W = weight of drill string (on moon)
 N = normal force against drill string
 f = friction force against drill string
 μ = coefficient of friction (soil against drill string) = 2.0
 T = torque from friction force
 P = power required to rotate drill string

$$N = W \sin (15^\circ)$$

$$f = \mu N$$

$$T = f \times r$$

$$P = T \times \omega$$

$$W = 2448 \text{ N (on moon)}$$

$$N = (2448 \text{ N})(0.2588) = 633.5 \text{ N}$$

$$f = (2)(633.5 \text{ N}) = 1267 \text{ N}$$

$$T = (1267 \text{ N})(.095\text{m}/2) = 60.2 \text{ N}\cdot\text{m}$$

$$P = (60.2 \text{ N}\cdot\text{m})(31.42 \text{ rad/sec}) = \underline{1.89 \text{ kW}}$$

Drill String Analysis

Weight of soil cuttings:

$$\text{Volume} = \frac{\pi}{4}(0.1\text{m})^2(5\text{m}) = 0.0393\text{m}^3 \quad \text{for a single drill string section}$$

$$\text{Density of cuttings} = \frac{3\text{ g}}{\text{cm}^3} \times \frac{(100\text{ cm})^3}{1\text{ m}^3} \times \frac{\text{kg}}{1000\text{ g}} = 3000\text{ kg/m}^3$$

$$\begin{aligned}\text{Mass of cuttings in 5 meter section} &= 3000\text{ kg/m}^3 \times 0.0393\text{ m}^3 = 118\text{ kg} \\ &= 193\text{ N (on moon)}\end{aligned}$$

Case I: Drill string yields in tension

This checks for yield in the drill string when 25 coupled sections are lifted from the bottom of a drilling hole. The total tension force is the sum total of the weight of the drill strings and a 5 meter section full of cuttings, which should be the maximum amount of cuttings in the entire drill string at any one time.

$$\text{weight of 25 drill strings} = 1782\text{ kg} = 2914\text{ N (on moon)}$$

$$\text{tension force} = 2914 + 193 = 3107\text{ N}$$

$$\text{drill string cross sectional area} = \frac{\pi}{4}(.095^2 - .085^2) = 1.414 \times 10^{-3}\text{ m}^2$$

$$\sigma = \frac{3107\text{ N}}{1.414 \times 10^{-3}\text{ m}^2} = 2.2\text{ MPa}$$

$$\text{safety factor } n = \frac{\sigma_y}{\sigma} = \frac{241\text{ MPa}}{2.2\text{ MPa}} = 109 \quad \sigma_y = 241\text{ MPa (303 Stainless, weak steel)}$$

Case II: Drill String yields in shear

T = applied torque

τ = shear stress

J = polar moment of inertia

$$T_{\max} = T_{\text{bit}} + T_{\text{sleeve friction}}$$

$$\tau_y = 0.5\sigma_y$$

$$\tau_{\max} = \frac{T \cdot r_o}{J}$$

$$J = \frac{\pi}{32} (d_o^4 - d_i^4)$$

$$T_{\max} = 110 \text{ N-m} + 60.2 \text{ N-m} = 172.2 \text{ N-m}$$

$$\tau_y = (0.5)(241 \text{ MPa}) = 120.5 \text{ MPa}$$

$$J = \frac{\pi}{32} (.095^4 - .085^4) = 2.87 \times 10^{-6} \text{ m}^4$$

$$\tau_{\max} = \frac{(170.2 \text{ N-m})(.048 \text{ m})}{2.87 \times 10^{-6} \text{ m}^4} = 2.85 \text{ MPa}$$

$$\text{safety factor } n = \frac{120.5 \text{ MPa}}{2.85 \text{ MPa}} = 42.3$$

Case III: Drill String Buckles as a column under purely axial compressive force

Assume column ends are pinned. Check for buckling of 1 drill string section (5 meters) while above ground, just as boring into lunar surface begins. After boring into lunar surface, buckling is unlikely due to close fit within hole, thus only a 5 meter length is analyzed.

L = length of section = 5m

E = modulus of steel = 207 GPa

I = area moment of inertia

P_{cr} = critical load, maximum allowable load to prevent buckling.

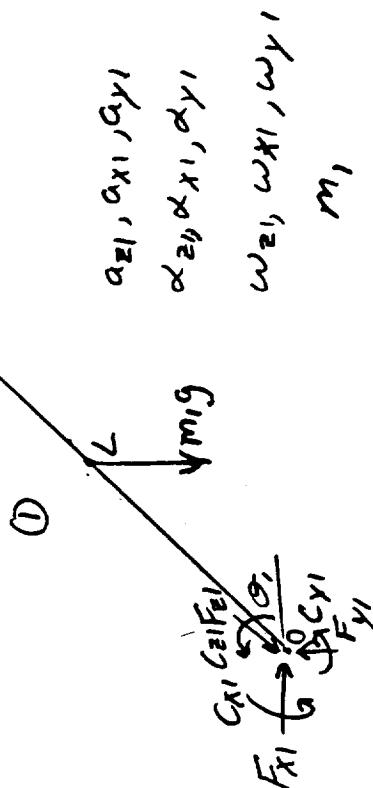
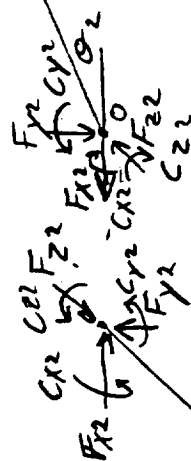
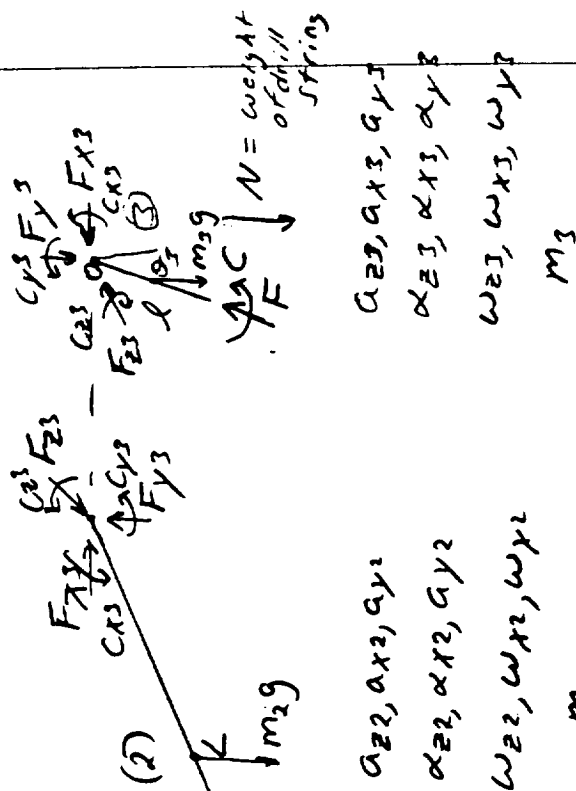
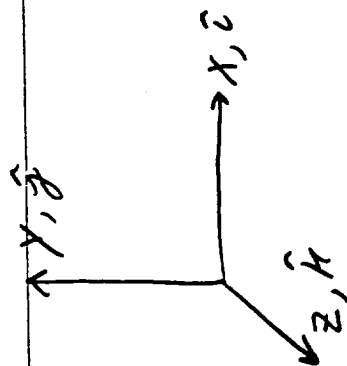
4400 N maximum applied downward force on drill string

$$I = \frac{\pi}{64}(d_o^4 - d_i^4) = \frac{\pi}{64}(.095^4 - .085^4) = 1.43 \times 10^{-6} \text{ m}^4$$

$$P_{cr} = \frac{\pi^2(207 \times 10^9 \text{ N/m}^2)(1.43 \times 10^{-6} \text{ m}^4)}{(5\text{m})^2} = 1.17 \times 10^5 \text{ N}$$

$$\text{safety factor } n = \frac{1.17 \times 10^5 \text{ N}}{4400 \text{ N}} = 26.6$$

Analysis of Robot Arm



Forces and Moments at Pin 1

$$\sum F = (F_{x1} + F_{x2})\hat{i} + (F_{y1} + F_{y2} - m_1 g)\hat{j} + (F_{z1} + F_{z2})\hat{k} = m_1(a_{x1}\hat{i} + a_{y1}\hat{j} + a_{z1}\hat{k})$$

$$\bullet F_{x1} = m_1 a_{x1} - F_{x2}$$

$$\bullet F_{y1} = m_1 a_{y1} - F_{y2} + m_1 g$$

$$\bullet F_{z1} = m_1 a_{z1} - F_{z2}$$

$$\bullet C_{x1} = -C_{x2} - F_{z2}L \sin \Theta_1 + I_{xx}\alpha_{x1} - (I_{yy} - I_{zz})\omega_{y1}\omega_{z1}$$

$$\bullet C_{y1} = -C_{y2} + F_{z2}L \cos \Theta_1 + I_{yy}\alpha_{y1} - (I_{zz} - I_{xx})\omega_{z1}\omega_{y1}$$

$$\bullet C_{z1} = -C_{z2} + m_1 g(l/2) \cos \Theta_1 - F_{y2} \cos \Theta_1 + F_{x2} \sin \Theta_1 + I_{zz}\alpha_{z1} - I_{yy}\omega_{x1}\omega_{y1}$$

Forces and Moments at Pin 2

$$\sum F = (F_{x3} - F_{x2})\hat{i} + (F_{y3} - m_2 g - F_{y2})\hat{j} + (F_{z3} - F_{z2})\hat{k} = m_2(a_{x2}\hat{i} + a_{y2}\hat{j} + a_{z2}\hat{k})$$

$$\bullet F_{x2} = F_{x3} - m_2 a_{x2}$$

$$\bullet F_{y2} = F_{y3} - m_2 g - m_2 a_{y2}$$

$$\bullet F_{z2} = F_{z3} - m_2 a_{z2}$$

$$\bullet C_{x2} = C_{x3} - I_{xx}\alpha_{x2} + (I_{yy} - I_{zz})\omega_{y2}\omega_{z2} + F_{z2}L \sin \Theta_2$$

$$\bullet C_{y2} = C_{y3} - I_{yy}\alpha_{y2} + (I_{zz} - I_{xx})\omega_{y2}\omega_{z2} + F_{z2}L \sin \Theta_2$$

$$\bullet C_{z2} = C_{z3} - m_2 g(l/2) \cos \Theta_2 - I_{zz}\alpha_{z2} - (I_{xx} - I_{yy})\omega_{x2}\omega_{y2} + F_{y3} \cos \Theta_2 - F_{y3} \sin \Theta_2$$

Forces and Moments at Pin 3

$$\sum F = F(\sin \Theta_3 \hat{i} + \cos \Theta_3 \hat{j}) - N \hat{j} - m_3 g \hat{j} - F_{z3} \hat{k} - F_{x3} \hat{i} - F_{y3} \hat{j} = m_3(a_{z3} \hat{k} + a_{x3} \hat{i} + a_{y3} \hat{j})$$

$$\bullet F_{x3} = F \sin \Theta_3 - m_3 a_{x3}$$

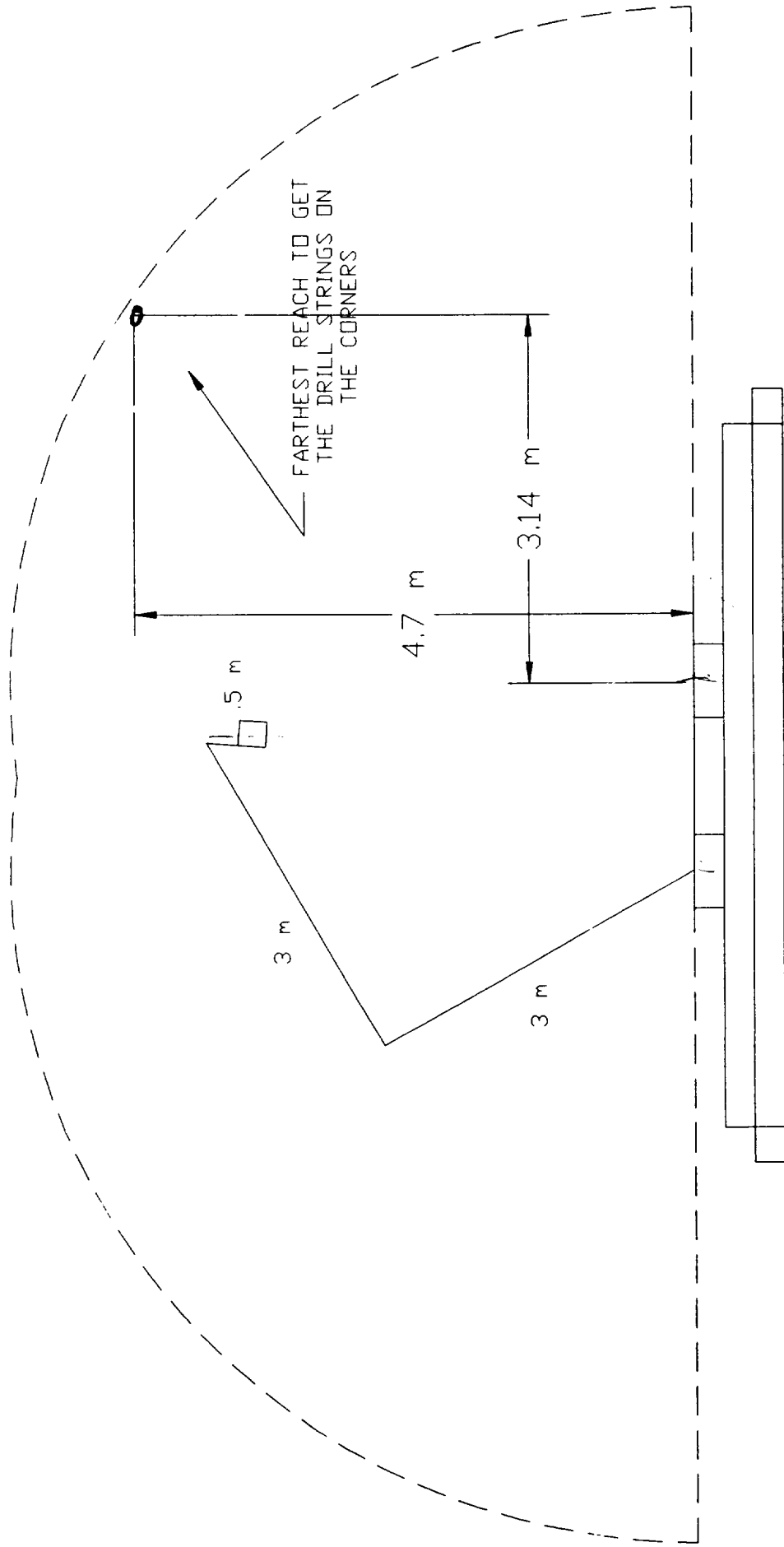
$$\bullet F_{y3} = F \cos \Theta_3 - N - m_3 g - m_3 a_{y3}$$

$$\bullet F_{z3} = -m_3 a_{z3}$$

$$\bullet C_{x3} = C \sin \Theta_3 - I_{xx} \alpha_{x3} + (I_{yy} - I_{zz}) \omega_{y3} \omega_{z3}$$

$$\bullet C_{y3} = C \cos \Theta_3 - I_{yy} \alpha_{y3} + (I_{zz} - I_{xx}) \omega_{z3} \omega_{x3}$$

$$\bullet C_{z3} = m_3 g (l/2) \sin \Theta_3$$



SURFACE

Constants

$h =$	0.30	m	$\rho =$	2700.00	kg/m ³			
$h' =$	0.28	m						
$t =$	0.01	m						
x-Dir			y-Dir			z-Dir		
a	0.00	m/sec ²	a	0.00	m/sec ²	a	0.00	m/sec ²
α	0.00	rad/sec ²	α	0.00	rad/sec ²	α	0.00	rad/sec ²
ω	0.00	rad/sec	ω	0.00	rad/sec	ω	0.00	rad/sec
$F =$	4400.00	N	$C =$	110.00	N*m	$\theta_3 =$	0.26	rad
$N =$	0.00	N	$g =$	1.63	m/sec ²	$\theta_2 =$	0.70	rad
						$\theta_1 =$	2.80	rad

1st Arm Pin

$L =$	3.00	m						
$m =$	120.19	kg	$F_x =$	1139.66	N	$C_x =$	28.49	N*m
$I_{xx} =$	1.19	kg*m ²	$F_y =$	3824.58	N	$C_y =$	106.25	N*m
$I_{yy} =$	0.59	kg*m ²	$F_z =$	0.00	N	$C_z =$	3607.28	N*m
$I_{zz} =$	0.59	kg*m ²						

2nd Arm Pin

$L =$	3.00	m						
$m =$	120.19	kg	$F_x =$	1139.66	N	$C_x =$	28.49	N*m
$I_{xx} =$	1.19	kg*m ²	$F_y =$	4020.86	N	$C_y =$	106.25	N*m
$I_{yy} =$	0.59	kg*m ²	$F_z =$	0.00	N	$C_z =$	285.63	N*m
$I_{zz} =$	0.59	kg*m ²						

3rd Arm Pin

$L =$	0.50	m						
$m =$	20.03	kg	$F_x =$	1139.66	N	$C_x =$	28.49	N*m
$I_{xx} =$	0.20	kg*m ²	$F_y =$	4217.13	N	$C_y =$	106.25	N*m
$I_{yy} =$	0.10	kg*m ²	$F_z =$	0.00	N	$C_z =$	2.12	N*m
$I_{zz} =$	0.10	kg*m ²						

Momements of area**1st Bar**

$I_x =$	0.00	m ⁴
$I_y =$	4.80	m ⁴
$I_z =$	4.80	m ⁴

2nd Bar

$I_x =$	0.00	m ⁴
$I_y =$	4.80	m ⁴
$I_z =$	4.80	m ⁴

3rd Bar

$I_x =$	0.00	m ⁴
$I_y =$	0.02	m ⁴
$I_z =$	0.02	m ⁴

Stress

x-Dir			x-Dir			x-Dir		
$\sigma =$	0.00	MPa	$\sigma =$	0.00	MPa	$\sigma =$	0.00	MPa
$\tau =$	0.08	MPa	$\tau =$	0.08	MPa	$\tau =$	0.08	MPa
y-Dir			y-Dir			y-Dir		
$\tau =$	226.29	MPa	$\tau =$	18.84	MPa	$\tau =$	31.65	MPa
z-Dir			z-Dir			z-Dir		
$\tau =$	0.00	MPa	$\tau =$	0.00	MPa	$\tau =$	0.00	MPa

TABLE 2
STRESS ANALYSIS ON EACH LINK OF THE ROBOT ARM

Constants

$h =$	0.30	m	$\rho =$	2700.00	kg/m ³			
$h' =$	0.28	m						
$t =$	0.01	m						
x-Dir			y-Dir			z-Dir		
a	2.00	m/sec ²	a	2.00	m/sec ²	a	2.00	m/sec ²
α	2.00	rad/sec ²	α	2.00	rad/sec ²	α	2.00	rad/sec ²
ω	2.00	rad/sec	ω	2.00	rad/sec	ω	2.00	rad/sec
$F =$	0.00	N	$C =$	0.00	N*m	$\theta_3 =$	0.26	rad
$N =$	116.00	N	$g =$	1.63	m/sec ²	$\theta_2 =$	0.70	rad
						$\theta_1 =$	2.80	rad

1st Arm Pin

$L =$	3.00	m						
$m =$	120.19	kg	$F_x =$	40.06	N	$C_x =$	274.39	N*m
$I_{xx} =$	1.19	kg*m ²	$F_y =$	188.78	N	$C_y =$	657.58	N*m
$I_{yy} =$	0.59	kg*m ²	$F_z =$	40.06	N	$C_z =$	166.56	N*m
$I_{zz} =$	0.59	kg*m ²						

2nd Arm Pin

$L =$	3.00	m						
$m =$	120.19	kg	$F_x =$	200.32	N	$C_x =$	75.45	N*m
$I_{xx} =$	1.19	kg*m ²	$F_y =$	247.89	N	$C_y =$	94.90	N*m
$I_{yy} =$	0.59	kg*m ²	$F_z =$	200.32	N	$C_z =$	188.64	N*m
$I_{zz} =$	0.59	kg*m ²						

3rd Arm Pin

$L =$	0.50	m						
$m =$	20.03	kg	$F_x =$	40.06	N	$C_x =$	0.40	N*m
$I_{xx} =$	0.20	kg*m ²	$F_y =$	188.78	N	$C_y =$	0.59	N*m
$I_{yy} =$	0.10	kg*m ²	$F_z =$	40.06	N	$C_z =$	17.34	N*m
$I_{zz} =$	0.10	kg*m ²						

Momements of area**1st Bar**

$I_x =$	0.00	m ⁴
$I_y =$	4.80	m ⁴
$I_z =$	4.80	m ⁴

2nd Bar

$I_x =$	0.00	m ⁴
$I_y =$	4.80	m ⁴
$I_z =$	4.80	m ⁴

3rd Bar

$I_x =$	0.00	m ⁴
$I_y =$	0.02	m ⁴
$I_z =$	0.02	m ⁴

Stress

x-Dir			x-Dir			x-Dir		
$\sigma =$	0.00	MPa	$\sigma =$	0.00	MPa	$\sigma =$	0.00	MPa
$\tau =$	0.73	MPa	$\tau =$	0.20	MPa	$\tau =$	0.00	MPa
y-Dir			y-Dir			y-Dir		
$\tau =$	0.52	MPa	$\tau =$	0.77	MPa	$\tau =$	11.60	MPa
z-Dir			z-Dir			z-Dir		
$\tau =$	0.43	MPa	$\tau =$	0.31	MPa	$\tau =$	13.47	MPa

TABLE 2
STRESS ANALYSIS ON EACH LINK OF THE ROBOT ARM

4.1 ABSTRACT

Our job is to create a system that would automatically mine ore from beneath the moon's surface. Our system is a version of an auger. An auger is fitted with a tube that prevents material spillage. The auger is split into seven sections of 15 m length. This auger is driven by a 16 kW motor that connects with a special threaded fastener attachment. Two 1.5 kW motors that connect to the torque motor with cables provide the needed bearing force. In addition, a gripper is used to constrain the auger sections during connection/disconnection. A rack is used to store the section when not in use.

4.2 INTRODUCTION

Extraterrestrial bodies--planets, moons, and asteroids--are available as possible sites for permanent bases in space. These bases would offer a place to observe the universe free from the distortions of the earth's atmosphere, a staging site for further space exploration, or a place from which to exploit mineral resources of the host body.

The moon, our nearest celestial neighbor, is a promising site for the first extraterrestrial support base. The absence of an adequate atmosphere on the moon, and the presence of cosmic radiation and micrometeor hazards, suggest that permanent installations will have to be constructed below the surface for adequate protection. Normal terrestrial excavation equipment, which rely upon gravity to provide the reaction forces for efficient operations are very heavy and thus too costly to take into space. In addition, the use of space suits required for work outside a controlled environment will make such efforts by man very difficult and inefficient. Automated mining operations have been developed, however, to excavate a small amount of ore deposited under the moon surface.

4.3 GOALS AND CONSTRAINTS

The prototype lunar mining system should perform economically and dependably from startup to decommission. The system should meet the following goals and constraints. However, some of these goals and constraints may prove to be conflicting, in which case compromises and tradeoffs will have to be made.

GOALS :

1. It must accept and produce the volume of $9 \text{ m}^3/\text{hr}$.
2. It must be able to operate in harsh moon environment.
3. The system must be fully automated.
4. The equipment should be versatile and be used for deep or shallow mining.

CONSTRAINTS :

1. The entire system's power assumption is limited to 25 kW.

2. The system must withstand maximum load of 6000 kg.
3. Must be able to withstand wide range of temperatures between -200 °C and 125 °C.
4. It must be able to operate in atmosphere-less environment.
5. It must be able to withstand constant cosmic radiation.
6. Must be able to mine at maximum depth of 100 m.

4.4 ORDER OF OPERATIONS (see fig. 4.1)

The mining process begins after the drill team finishes the exploration on the site. This exploration pinpoints where the ore is for our mining equipment to bring to the surface. The first step is to fragment the ore into particles small enough for an auger mining system to easily transport to the surface. This is handled by explosives placed into the drill team's exploration holes.

After the ore is fragmented the trolley, motor, gripper assembly attached to the material handling team's I-beam connects to the stuart platform. This is accomplished by rolling the trolley from the storage I-beam to the platform I-beam. The gripper must be dropped into place before any sections can be attached. The platform lowers to the moon's surface directly above the spot where mining is required. The gripper's solenoid locking / unlocking mechanism activates to separate it from the motor. The gripper remains on the moon's surface until mining is complete.

In order to supply the downward force necessary for mining, bearing cables are attached. The cable is housed in a winch motor and spool connects to the 10 m ring on the structure. The cable is also threaded through a pulley which is 0.5 m off the moon's surface. A hook is on the end of the cable which is used to attach to the I-hooks on the trolley. The platform lowers the trolley / motor assembly into place to connect to both of the bearing force cables.

The mining sections are stored in a rack assembly which sits on the moon's surface and connects to the lowest ring on the structure. There are seven 15 m long sections which lie in the rack at a 15° angle. The first section has an added drill bit for mining into the ground. The platform moves the motor into position above the end of the first bit. The motor has a threaded fastener which screws into the section. The rack is designed to resist the turning of the bit to ensure proper fastening. After the motor screws into the bit the proper amount, three solenoid operated clamps prevent the bit from disengaging from the motor during rotation.

After connecting, the first bit moves over the hole for mining. The bearing force cables pull the rotating assembly down into the ground through the opening in the gripper. When the first bit is fully in the ground it is the responsibility of the gripper to secure it while the motor detaches to get the next section. The

gripper's solenoid locking device activates and the bit secures. The motor then disengages from the bit in the exact opposite order from which it attaches.

The motor then attaches to the next mining section. Connection procedures are exactly the same as the previous bit as well as all remaining bits. Once connected the bit is placed directly above the bit being held by the gripper on the moon's surface. The bit to bit connection is another threaded fastener exactly the same as the motor to bit connection. However, since the gripper supplies the necessary resistance to torque, the three clamps that the motor to bit connection needs is not needed with the bit to bit connection.

After connecting the two sections the downward mining process resumes. The number of sections added for the entire project depends on the location and depth of the ore found by the drilling team. The total possible mining depth is 100 m.

The material is brought to the surface through a series of steps. The first trip downward yields no material at the surface. At this point the auger fills up with the ore and overburden. After the assembly reaches its maximum depth it raises back up to allow material to fill in the open crevasse. Since the material is fragmented by previous explosions it only requires light brushings by the drill bit on its way up to fall down and fill the hole. The following trips down forces the material in the auger out of the top and replace it with more material.

This process continues until a desirable amount of the ore is mined. Dismantling of the system is similar to connecting. The gripper holds the lower sections into place while the top section is removed and placed into the rack. The bearing force cables are then removed and are returned to their normal position, flush against the pulley. Finally the gripper reconnects to the motor and the trolley, motor, gripper assembly returns to its spot on the material handling's storage I-beam.

The system has the following components :

- Explosives
- Trolley
- Torque Motor
- Auger Sections
- Gripper
- Rack
- Bearing Cables

The following sections contain the in-depth descriptions of each component. Calculations for each can be found in appendix I. Dimensions can be found in the drawings of appendix II.

4.5 DESIGN DESCRIPTION

4.5.1 EXPLOSIVES (see fig. 4.2)

4.5.1.1 CONCEPT

Blasting is the rapid decomposition of a small volume of explosive within a mass of rock. Detonation of the explosive forms the gaseous products of decomposition with such rapidity and in such volumes that great and almost instantaneous pressure is exerted on the rock mass. Because this explosive energy is confined, the rock mass is fractured.

4.5.1.2 MATERIAL

Chemical high explosives are hazardous materials that require specific handling and shipping procedures when used in our terrestrial environment. Many of these compounds are not suitable for space applications. For example, although ANFO is commonly used on earth, it is not suitable for space. Requirements for selecting explosives for use in space are listed below:

1. Safe to handle and use
2. Maintain properties at temperatures of -60 to 127 °C in an insulated package for 1 year
3. Exhibit low volatility under pressures of 10^{-12} to 10^{-14} torr, and under the above temperature conditions
4. Capable of being pressed to high densities and then machined to fit a grenade case
5. Capable of withstanding expected shock and vibration during launch, lunar landing, and emplacement
6. Develop the explosive energy of TNT
7. Maintain properties after exposure to anticipated radiation levels
8. Initiate reliably with the available initiation system
9. Reasonable material availability

Some candidate explosives which satisfy these requirements are listed in the appendix (see fig. 4.2).

4.5.2 TROLLEY (see fig. 4.3)

4.5.2.1 CONCEPT

The trolley is a device used to transfer the motor from storage to operations areas. It uses an I-beam as a track. There are two I-beams that are used. The first is a storage I-beam which connects to the structure out of the way of the operation areas. The second permanently attaches to the bottom of the stuart platform. This I-beam is used during all operational phases of the mining process. The

trolley utilize rollers to translate it from one I-beam to the next.

4.5.2.2 MATERIAL

The bulk of the trolley is made from aluminum. This allows the trolley to retain its strength while minimizing the mass. The entire assembly weighs only 12 kg. The rollers are constructed of hard rubber.

4.5.2.3 DESIGN

1. The main body of the trolley is "C" shaped to interface with the I-beam. The dimensions are sufficient enough fit around the I-beam while large enough to support the motor. The bottom of the trolley is 32 cm x 16 cm. The entire piece is 6 cm tall. (Refer to Engineering Drawing T, in the Appendix). The aluminum is 1 cm thick to support the weight of the rest of the mining equipment. Bolts attach the trolley to the motor. Two I-hooks also attach to the main body. These I-hooks are made from aluminum rods 1 cm in diameter. The inner diameter of the hole is 1.2 cm. These hooks are used to connect the trolley to the bearing force cables which supply the downward pull needed to mine into the ground.

2. There are 7 rollers on each side of the top of the trolley. They span the entire 32 cm. Each roller is 3 cm in diameter with a 1 cm bolt as its axle. Spacing between the rollers is 7 mm with 0.5 mm space left at the ends.

4.5.3 MOTOR (see fig. 4.4)

4.5.3.1 CONCEPT

The motor is capable of providing enough torque to turn a fully loaded auger. It is well under the 25 kW power limit to allow other operations to be performed at the same time. Connectors should allow quick connections and disconnections to all the sections. A method is also incorporated to allow the gripper to connect during periods of storage.

4.5.3.2 MATERIAL

The motor is a brush servo motor. This is chosen to provide the massive torque required to rotate a 100 m auger full of ore. The bolts used to connect the motor to the trolley and the threaded attachment used to connect the motor to the auger are aluminum due to its high strength to weight ratio.

4.5.3.3 DESIGN

1. It is assumed that material handling consumes a maximum of 6 kW of power during mining, which leaves our system with 19 kW of power. The DC brush servo motor requires 16 kW of power to supply its maximum torque of 18,067

Nm. At this torque, it is capable of rotating at 368.5 rpm. The motor is cylindrical in shape. Dimensions are from an Electro-Craft Servo Systems catalog as:

diameter => 17.8 cm

length => 50 cm

2. The connection to the trolley is a flat plate 30 cm x 30 cm. This is 2 cm thick to support the total system weight. The eight bolts holding the trolley plate to the motor plate are 2.54 cm in diameter.

3. The connect for the gripper is similar to the connections used by the sections for the gripper. The motor casing has a hexagonal shaped collar 20 cm from the top. This is 7.5 cm wide. The purpose of the hexagon is to keep the clamps of the gripper from sliding.

4. The connection for the sections is a threaded fastener with three additional clamps. The threaded fastener is 1.12 m long and 3 cm in diameter. The threads are 1 mm thick and the pitch is 2 mm. There are 561 threads on the 2.4 cm core. This threaded fastener screws to the top of the sections. To prevent the fasteners from becoming unconnected during forward and reverse running, 3 clamps are used. These clamps are oriented around the threaded fastener. Each clamp is 1.2 cm in diameter and 5 cm in length. The operation of the clamps is a solenoid within the motor base.

4.5.4 AUGER SECTIONS (see fig. 4.5)

4.5.4.1 CONCEPT

The sections are continuous auger style designs. They are dimensioned to ensure that they fit within the maximum height of 20 m for the stuart platform as determined by the materials handling team. There are enough of them to fulfill the 100 m depth of mining requirement. The connections between the sections and the motor are easily performed with rotation.

4.5.4.2 MATERIAL

The material for the shafts is made from a 60% boron fiber and 40% polyethersulfone matrix to meet strength and weight ratios. All auger parts are made from this same material. The material choices of composites are chosen because they can provide a specific tensile strength that is approximately four to six times greater than steel or aluminum, the composites can provide a specific modulus that is three and a half to five times greater than steel or aluminum, and the fatigue endurance limit is much higher than for steel or aluminum. Also toughened composites can give impact energies significantly higher than aluminum alloys. Design flexibility of the composites is greater and can allow for physical property directionality in parts where desired. The potential for

corrosion is significantly reduced. And, the composite materials can eliminate many joints and can be fastened by simplified methods, thus eliminating both structural weaknesses and manufacturing costs. The shaft is manufactured using filament winding for multi-axial and radial strengths. In order to fight the Moon's cosmic radiation, an anti-radiation treatment is applied to the composite auger system.

4.5.4.3 DESIGN

1. There are seven total sections, each consists of 2 main parts, a threaded core and a tube. They are both 15 m long but are offset from each other. The core starts 10 cm higher than the tube and also ends 10 cm higher. This gives us the necessary clearance for the ore to spill out of the top of the auger and onto the moon's surface. The core is an auger with 1 cm thick flights 4 cm apart. The helical angle of the flights is 37.5° .
2. The core diameter is 5.39 cm. The top of the core has a threaded hole 3 cm in diameter, 1.12 m in length, a thread thickness of 1 mm, and a pitch of 2 mm. There are also three small indentations around the threaded hole for the anti rotation clamps of the motor. The threaded shaft serves as a connecting element from the shaft to either the motor or the other shafts.
3. The tube connects to the end of the threads. Its outer diameter is 17.82 cm while the inner diameter is 13.82 cm. This results in a thickness of 2 cm. The tube serves the purpose of keeping the ore on the threads.
4. The tube has a circular cross section for the entire length on its inner wall. The outer wall is circular except at the connection for the gripper. This connection is a hexagon inscribed in the outer diameter of the tube. The hexagon starts 7.5 cm from the top and is 7.5 cm long. The gripper has matching connections and fits over it and locks it tight. This prevents it from both falling and rotating while the other sections are being removed.
5. The last 6 sections are all identical in design. The first section includes a bit to promote mining into the ground. For this section the core and auger do not stop 10 cm before the end of the tube. Instead it continues at its same diameter until it is flushes with the end of the tube. The end of the thread is a blade used to scoop the material into the auger. The core then decreases in diameter extending into a point. This extension is 15 cm long. This point provides an initial hole for the auger to pass through. The end of the tube has spikes that are used for grinding into the ore. These spikes are 3 cm long and orients at various angles to dig into the material.

4.5.5 GRIPPER (see fig. 4.6)

4.5.5.1 CONCEPT

Gripper is used as a means to hold the drill sections in place while the motor disengages when retrieving the drill from underground. Due to height limitations of the dome structure, the drill is divided up in small sections. The gripper first attaches to the motor. The gripper must be dropped on top of the hole where mining is to take place before any sections can be attached. The platform lowers to the moon's surface and disengages from the motor. The drill shaft lowers through the gripper's hole. The gripper remains on the moon's surface until mining is complete. Once the mining operation is over and it is time to retrieve the drill shaft, one section of the drill is taken out at a time. The stuart platform raises in order to clear the section that needs to be removed. The gripper provides resistance against rotation of rest of the shaft while the top section disengages from rest of the shaft. Once the top section disengages, the platform puts the section in the rack and returns to the gripper for the next section. The platform lowers itself and the motor engages on to the next section, and the process repeats until the last section is removed from the gripper. The gripper's parts consist of main body, clamps, electro-magnets, battery compartment, track, and spikes. See appendix

4.5.5.2 MATERIAL

Steel is chosen due to its relatively high mechanical properties. Also it chosen for its heavy density. The high density is required in order to provide the downward force so that the gripper stays in place while the motor disengages the top drill section from rest of the body.

4.5.5.3 DESIGN

1. The main body is manufactured in solid steel in a cone shape. Middle sections is bored through in cylindrical shape in order for the drill shaft to slide up and down, and also rotate. The cone shape is chosen so that the material that is brought up to the surface by the Archemidian screw can be poured out from the gripper and slide down the gripper's sloped sides. About 77.9 cm above the bottom of the gripper, a rectangular tube is cut out to contain two clamps which slide back and forth.

2. A solid steel block's center is cut out in hexagonal shape which resembles the dimensions of the hexagonal part of the drill shaft. The block is then cut in half to form two clamps which can fit around the hexagonal part of the drill shaft. The clamps are then placed in the main body where the rectangular tube shape is cut out. The clamps are able to slide back and forth in the track. Before the operation the gripper hangs on to the motor. The motor, like the drill shaft, has a hexagonal part on the casing where the clamps are utilized for grip. When the

gripper lowers to the surface, the clamps disengage and the gripper disconnects from the motor. The clamps are in disengaged position until they are utilized for the retrieval of the drill shaft. When the platform raises to bring the top section out of the gripper, the clamps engage and they fit around the hexagonal part of the second drill shaft section. The hexagonal part on the drill section is longer than the clamps' height in case the shaft does not line up perfectly. The weight of the shaft is held up by the clamps. The tight fitting of the clamps which are sitting inside of the track provides the resistance necessary to turn and disengage the drill section that protrudes out of the gripper. The clamps stay in place until the motor returns, after putting away the section that is removed in the rack, for the next section. Once the motor engages on to the next section, while the clamps provides the resistance against the torque, the clamps disengage. The freed drill section then pulls up out of the gripper like its previous section. The clamps then engage again, attaching themselves on to the next drill section. The process repeats until the last section is removed from the gripper.

3. Electro-magnets are used to engage and disengage the clamps. They attach to the clamps' sections where the two clamps touch each other during engagement. When the opposite charges are turned on, the clamps attracts to each other, causing them to slide down the track and cling together. For disengagement, the same charges are provided, causing the clamps to push each other apart. The electro-magnets get their current from the battery located in the main body.

4. Battery compartment located in the main body of the gripper contains a battery. The battery compartment can be easily accessed from outside through the service door located on the sloped side of the main body. The battery gets charged by the motor. The battery provides the current necessary to operate the clamps. The battery compartment also contains a control device which provides different charged currents to engage and disengage the clamps.

5. Track is nothing but a rectangular tube cut out which has a slightly larger dimensions than the clamps for easy sliding. The clamps tight fitting in the track allows the clamps to hold the drill section in place when the motor disengages and also withstand the weight of the drill shaft. The track is manufactured with precision for smooth surface for the clamps to slide easily.

6. Spikes are designed to be attached to the main body when the main body is molded. The spikes are designed to provide a resistance against rotation of the motor by gripping on to the moon's soil to make sure the gripper itself does not rotate with the motor. The weight of the gripper, the shaft, and the material in the shaft will provide the normal force necessary to hold the gripper in place.

4.5.6 RACK (see fig. 4.7)

4.5.6.1 CONCEPT

The rack is used to store the drill sections when they are not being used. It has the capacity of storing up to seven sections at a time. The design can be used to accommodate more sections if prior arrangements are made on Earth. The whole system resembles a swing set from a playground. Basically two tubes are used to store the drill sections. Two sets of three tubes form a triangle for support. And, one last tube is used to hold those two triangle sections together. (See appendix)

4.5.6.2 MATERIAL

Aluminum has been chosen for its relatively good mechanical properties and light weight. The basic material for the design of the rack is eight hollow and one solid rectangular tubes of various lengths and shapes.

4.5.6.3 DESIGN

1. The top tube is used to hold top part of the seven drill sections. It is 3.68 m long and has seven indents in the shape of half hexagon. The indents are designed to accept the hexagonal part of the drill sections easily. Once the drill sections are in place, they will stay in place as far as rotations are concerned while the motor is in connection/disconnection mode. Even though the tube is not designed to provide a resistance against slippage of the drill sections, its hexagonal indents which has a smaller cross section than a cross section of the drill sections, will hold the sections from sliding down.

2. The bottom tube is used to hold bottom part of the seven drill sections. It also is 3.68 m long. Instead of hexagonal indents, it has seven half-circular indents. The gap of the half-circular indents are slightly bigger than the gap of the hexagonal indents. The half-circular indents are designed for a smooth fit for the circular drill sections. The deep indents are designed to accommodate a diameter of the sections plus a distance of a radii of the sections in order to provide a room in case of slippage. The bottom tube does not provide any resistance forces against rotation while the motor connects/disconnects the drill sections. It provides the resistance forces against the slippage of drill sections.

3. Two sections of triangle "legs" are designed to hold the top and bottom tubes of above apart and also hold the structure upright. The part that is connected to the top and bottom tubes are tilted at fifteen degrees. The angle is there so that the drill sections can be leaning against the rack instead of standing straight up, in which case the drill sections could fall back out through opening of the indents. The bottom part of the triangles can be used as it is or extended to be connected to the dome structure.

4. The last tube is used to provide a stability to the rack system. It is basically connected to the end of the "triangle legs" to keep them apart and in place. This is the only tube that is made of solid aluminum. Its purpose is to provide a downward force so that the rack would not tip over easily. This section can be totally eliminated in case of connecting the "triangle legs" to the dome structure.

4.5.7 BEARING CABLES (see fig. 4.8)

4.5.7.1 CONCEPT

Without the auger being self drilling, a problem of the moon's low gravity, there needs to be a device that supplies the bearing force that mining requires.

4.5.7.2 MATERIAL

The components of this system are made of steel since they are small but still need to be strong.

4.5.7.3 DESIGN

1. The motor will be a 1.5 kW Shunt wound 115 VDC gear motor. The gear ratio is 40 : 1 and the maximum speed is 43 rpm. It will be attached to the 10 m ring on the structure. Connected to it will be a spool 10 cm in diameter with a 8.4 cm base.
2. The steel cable will be 2 mm in diameter. It will be threaded through a pulley. The pulley will be 4 cm in diameter. On the end of the cable there will be a 3 cm diameter circular hook that is 285° closed. This hook is used to connect to the I-hook on the trolley.
3. The pulley is on a shaft that is 2.5 m long stuck 0.5 m into the ground. The pulley itself is mounted off center inside of a 10 cm x 10 cm box. The outside of the box is shaped like a pyramid. The hook is also shaped like a pyramid but hollow. Its purpose is to orient the hook when it is replaced.
4. The connections are made before the mining process begins. The platform lowers to allow the I-hooks on the trolley to connect to the cable's hook.

4.6 CONCLUSION

In conclusion, we feel that mining on the Moon in this manner would be quite feasible. This idea can be translated from paper to action with further research and study. We understand that we may run into conflicts when we actually put this idea in action. But with compromises and tradeoffs, we are sure we can overcome any conflicts. The goals set in section 4.3 have been met. Our system

is capable of producing $11.25 \text{ m}^3/\text{hr}$. This is a full 25% more material per hour than the goal volume of $9 \text{ m}^3/\text{hr}$ (based on the work needed to fill a truck with ore in 20 minutes). This is not efficient mining as compared to earth bound techniques. However, it is quite comparable to hand mining which is our goal for the remote location of the Moon. The system is designed to survive the harsh moon environment through proper material selection and treatment. All materials will survive the extreme temperature range and our composite material is protected from radiation. Also, the system is automated using control system except for the setup and remote monitoring parts. By being able to change the length of the drill shaft, the equipment is versatile and is used for deep or shallow mining. We are confident that this system is the first step to producing a fully automated mining system on the Moon.

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4.8 APPENDIX

4.8.1 TROLLEY (engineering drawing 3)

Most of the dimensions were taken from the given dimensions of the Materials Handling Team's I-beam. This was given as an Aluminum I-beam of designation W8x21. The mode of failure would be shearing between the trolley and the 1cm diameter bolt used as the axle for the rollers. This was examined as so:

$$\gamma = \frac{\tau}{G}, G_{Al} = 26.2 \text{ GPa}$$

$$\tau = \frac{F}{A_s}, F = (6000 \text{ kg})(1.67 \text{ m/s}^2), A_s = \frac{\pi}{4} (.01 \text{ m})^2$$

$$\tau = 12.76 \text{ MPa}$$

$$\gamma = 4.87 \text{ E} - 4$$

This value is extremely small. Therefore, there will be very little deflection and no shearing. The trolley will be safe.

4.8.2 MOTOR (engineering drawing 4)

The main problem for the motor was finding the needed torque and supplying this torque with the given power. The torque was given as follows:

$$T = T_{\text{auger}} + T_{\text{grinding}} + T_{\text{bit}}$$

Where the auger torque is that needed to lift the load, the grinding torque is that needed to turn through a densely packed earth, and the bit torque is the torque needed to turn the auger as it cuts into the ground. These were found as follows:

$$T_{\text{auger}} = \frac{F d_m}{2} \left(\frac{1 + \pi \mu d_m}{\pi d_m - \mu l} \right)$$

$$\mu = .119$$

$$d_m = D_c + \frac{1}{2} (D_i - D_c) = .10605 \text{ m (from sections)}$$

$$l = 5 \text{ cm (from sections)}$$

$$F = 4643.2 \text{ N (from sections)}$$

$$\therefore T_{\text{auger}} = 67 \text{ Nm}$$

Notice that the torque equation is one for a power screw. Therefore, the equation is oversimplifying the problem or it at least contains some inaccuracies. However, we could not obtain an equation for torque of an auger and assumed that the power screw equation

would be fairly accurate.

The next torque to be found was the grinding torque. This was done using two major assumptions. First, we assumed that densely packed material could be treated as a liquid with the exception that the rock would not have the same cohesion (one tenth that of a liquid). Secondly, we "unwrapped" the cylindrical tube so that it was a planar wall. This torque was found as:

$$F_{\text{resultant}} = .1\gamma \frac{bh^2}{2} = 936\text{kN}$$

$$F_{\text{friction}} = \mu F_{\text{resultant}} = 111.4\text{kN}$$

$$\therefore T_{\text{grinding}} = rF_{\text{friction}} = 17620\text{Nm}$$

The bit torque was found using the assumption that the friction coefficient between the bit and the earth would be 0.8. From Manufacturing International 1990 Volume IV, the thrust force needed for this bit would be 750 N. The bit torque is:

$$T_{\text{bit}} = \mu Fr = .8(750\text{N})(.1582\text{m}) = 380\text{Nm}$$

Therefore, the total torque that the motor must resist is 18067 Nm. We were given that our total amount of available power was to be 19 kW. Besides this main motor, our system also contains two smaller winch motors that are used to supply the bearing force needed during initial drilling. We set aside 1.5 kW for each of these two smaller motors. This left our main motor with 16 kW. Given this torque, we wanted to find our RPM and MRR (or material removal rate). These were found as follows: (Note that the RPM equation was received from Research Designs INC. in Tucker, Georgia).

$$\text{RPM} = \frac{416076P_{\text{kW}}}{T_{\text{Nm}}} = 368.5\text{rpm}$$

$$\text{MRR} = \left(\frac{\text{vol of material}}{\text{one revolution}}\right)(\text{RPM}) = \frac{\pi}{4} (.1382^2 - .0539^2) 04(66.7)(60) = 11.25 \text{ m}^3/\text{hr}$$

Although the value for RPM is fairly reasonable, the MRR is quite low. However, this is due to weight and power constraints that are beyond our control.

There will be a screw connection from the motor to the sections. This will be discussed in the section for sections.

4.8.3 SECTIONS (engineering drawing 5)

We began the design of the sections with some assumptions that were taken from a book (which is referenced in the text). First, it was assumed that the ratio of the inner diameter to the outer diameter would be 0.776 and that the ratio of the core diameter to the outer diameter would be 0.302. We then assumed that the threads would occupy one-fifth of the total volume of the tube. Thus, the ore could occupy up to four-fifths of the tube

volume. We wanted to maximize the total volume that could be mined, but we had to take weight limits into consideration. Our total weight limit was 6000 kg, and we assumed that the sections filled with ore could take 5000 kg. This would leave an additional 1000 kg for the remaining equipment.

We first looked at steel for the auger material. However, we found that even an empty tube would not remain within the bounds of the weight requirement. Further study showed that we should consider using some sort of composite material. We finally decided on a matrix of PES (polyestersulfone) with Boron fibers. The ratio was 60% Boron to 40% PES. Outside diameter of the tube was found as follows: (Note that it is purely coincidental that the density of the ore and the composite are the same).

$$\rho_{\text{composite}} = \rho_{\text{ore}} = 2000 \frac{\text{kg}}{\text{m}^3}$$

$$V = 78.54D_o^2$$

$$M = V\rho = 5000\text{kg}$$

$$\therefore D_o = 17.82\text{cm}$$

$$D_i = .776D_o = 13.82\text{cm}$$

$$D_c = .302D_o = 5.39\text{cm}$$

Using the one-fifth assumption for threads, we wanted to determine thread thickness. We first determined minimum thread thickness by examining the loaded thread as a beam fixed at both ends. The needed thickness was so small that we could make the final thickness to be any number. For ease of manufacturing, we chose 1cm. This leaves a 4 cm gap between threads and gives a helix angle of 43.5°.

The next thing that needed to be considered was the design of the fastener screws. The fastener screws are the connections between sections. It was assumed that the core of the auger would have a 3 cm diameter hole in which the fastener could attach. Therefore, the major diameter of the screw is 3 cm. From the ME 4180 textbook, a ratio of major diameter to core diameter was found to be 1.25. This gave a core diameter of 2.4 cm. We assumed a thread thickness of 1 mm. We then needed to determine the length of the threaded fastener based on the thickness of the thread (assuming the thread acts as a cantilevered beam):

$$v = \frac{qx^2}{24EI} (6L^2 - 4Lx + x^2) @ x = L \text{ for max. deflection}$$

$$E = .6E_{\text{boron}} + .4E_{\text{PES}} = .6(393\text{GPa}) + .4(2.62\text{GPa}) = 237\text{GPa}$$

$$\text{set } v = .01L = .00003\text{m}$$

$$q = \frac{(5000\text{kg}) \left(1.67 \frac{\text{m}}{\text{s}^2} \right)}{\frac{\pi (.03^2 - .024^2)}{4} n}, \text{ where } n \text{ is the number of threads}$$

We then solved for the number of threads and the length of the screw. We assumed a pitch of 2mm:

$$n = 561$$

$$L = 1.12\text{m}$$

For the hexagonal indentation, we simply planed off the minimum amount of material from the tube that would allow the hexagon to exist. Exact dimensions can be found in the engineering drawings. Similar assumptions were made for the holes in the core that are to be used for unscrewing one section from another.

4.8.4 BEARING FORCE CABLE SYSTEM (engineering drawing 8)

Most of this system was free designed. The only calculations that were needed involved finding the diameter of cable needed to withstand the bearing forces and the distance from the gripper to place the pulleys.

It was given (as stated earlier) that a bearing force of 750 N was required. Therefore, we needed to find the maximum force that the winch motors must pull through the cables. This occurs when the motor is at its lowest point of 1.1 m. From Bodine Electric Co., we found the maximum torque for a 1.5 kW motor to be 167 Nm. Assuming a winch spool of 8.5 cm in diameter, the maximum force that the winch can deliver is 1965 N. This defines the maximum distance from the edge of the trolley to the pulley to be x (where θ is the minimum angle):

$$F \sin \theta = 375\text{N}$$

$$\theta = 11^\circ = \tan^{-1} \frac{1.1}{x}$$

$$x = 4.73\text{m}$$

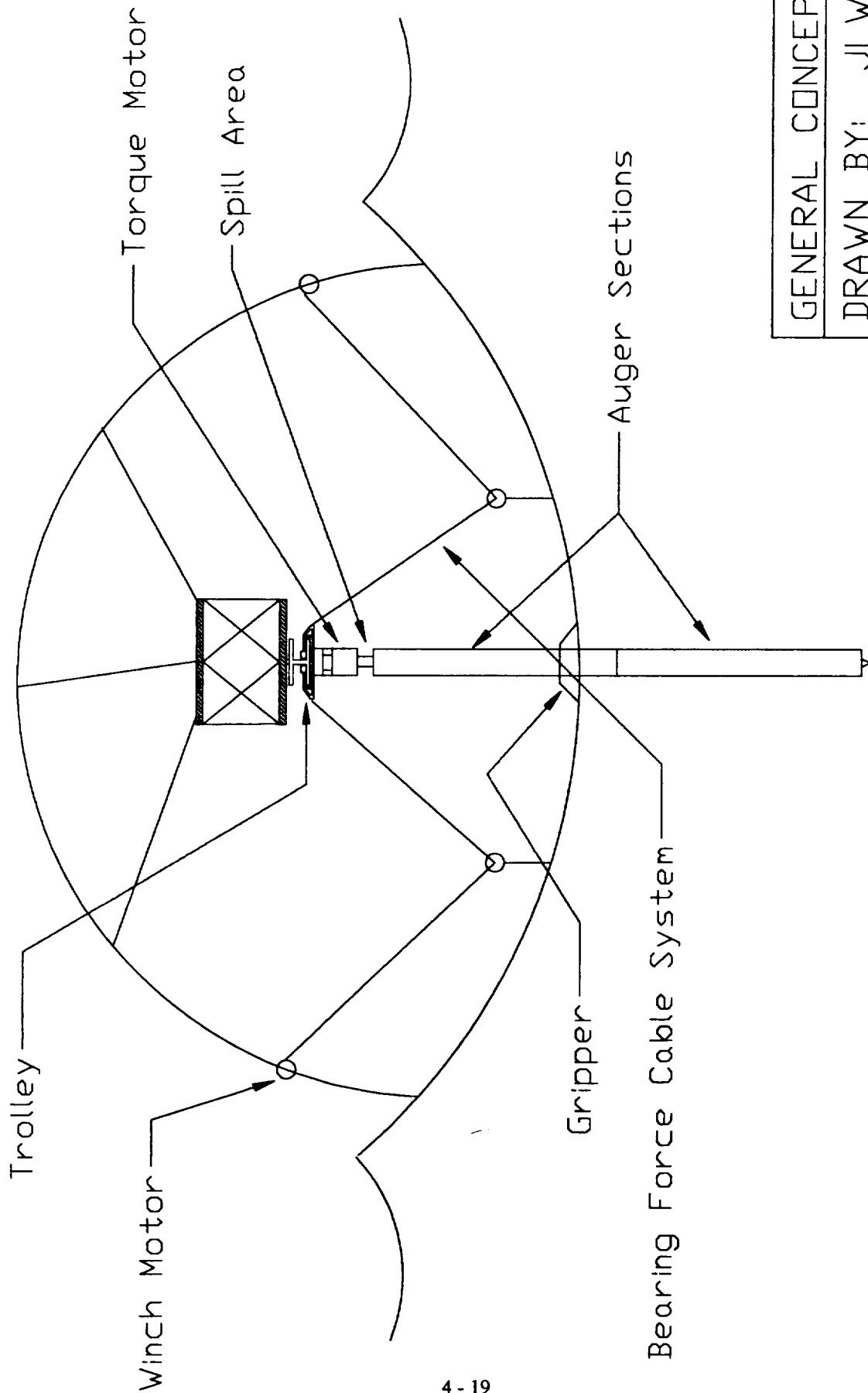
A simple stress analysis was performed in order to determine cable diameter. The material is assumed to be high strength steel:

$$\sigma \leq \frac{1}{2} \sigma_{TS}$$

$$\sigma_{TS} = 1\text{GPa}$$

$$\sigma = \frac{P}{A} = \frac{1965\text{N}}{\frac{\pi}{4} d^2}$$

$$\therefore d = 3.2\text{mm}$$

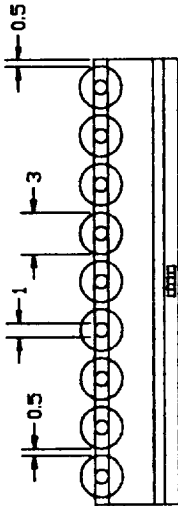
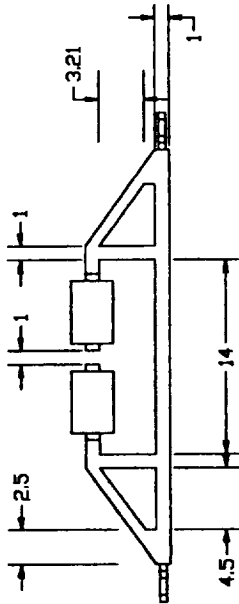
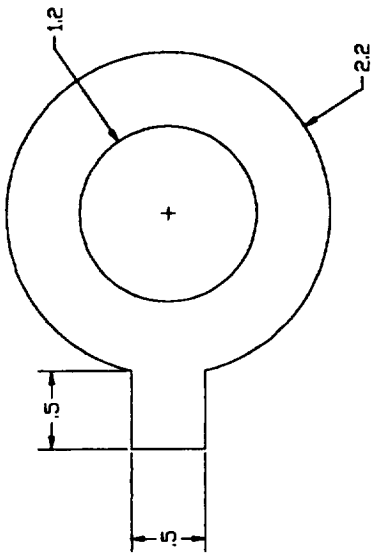
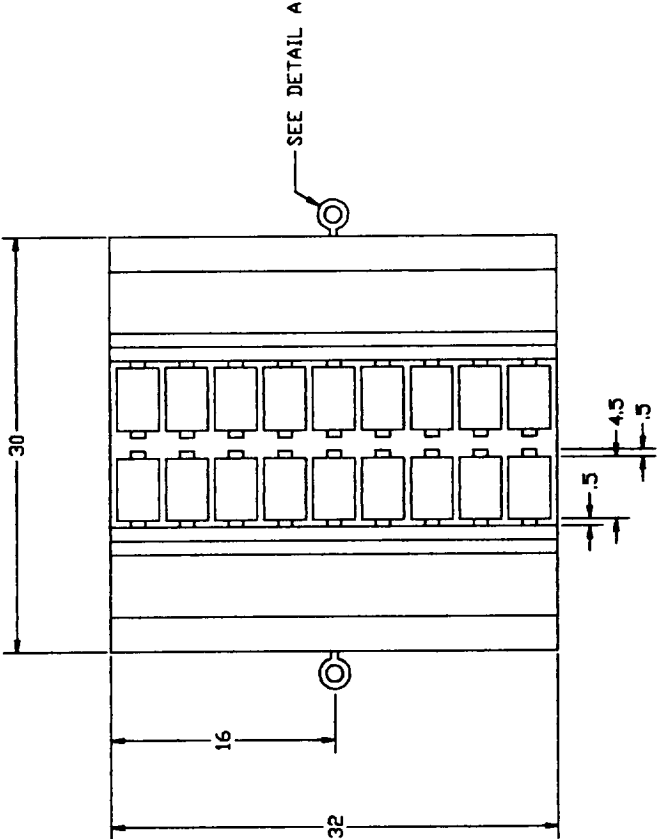


GENERAL CONCEPT
DRAWN BY: JLW
DATE: 3/10/94
ME 4182 MINING

4.8.6 Fig. 4.2 Explosives

Explosive Name	Composition Percentage by Weight	Detonation Density gm/cc	Pressure Kilobars	Energy Release cal/gm (cal/cc)	Impact Sensitivity cm	Thermal Stability cc/gm	Melting Point C
DATB	100 DATB	1.79	250	1000 (1790)	>177	<0.03	295
ALD	65 DATB 35 Al	2.00	~200	1500 (3000)	?	?	295
LX-04	85 HMX 15 Viton A	1.86	330	1320 (2460)	41-74	0.5	280
ALX	60 LX-04 40 Al	2.20	~270	2000 (4400)	?	?	280
HNS-II	90 HNS 10 Teflon	1.70	~160	1200 (2040)	63	Nil	318
ALH	68 HNS-II 32 Al	2.01	~120	1800 (3620)	?	?	318
MFH	28 Mg 62 Fe ₂ O ₃ 10 HNS-II	3.70	?	1100 (3900)	?	Nil	318
ALOX	53 Al 47 Liq. O ₂	1.95	?	3810 (7420)	?	--	--

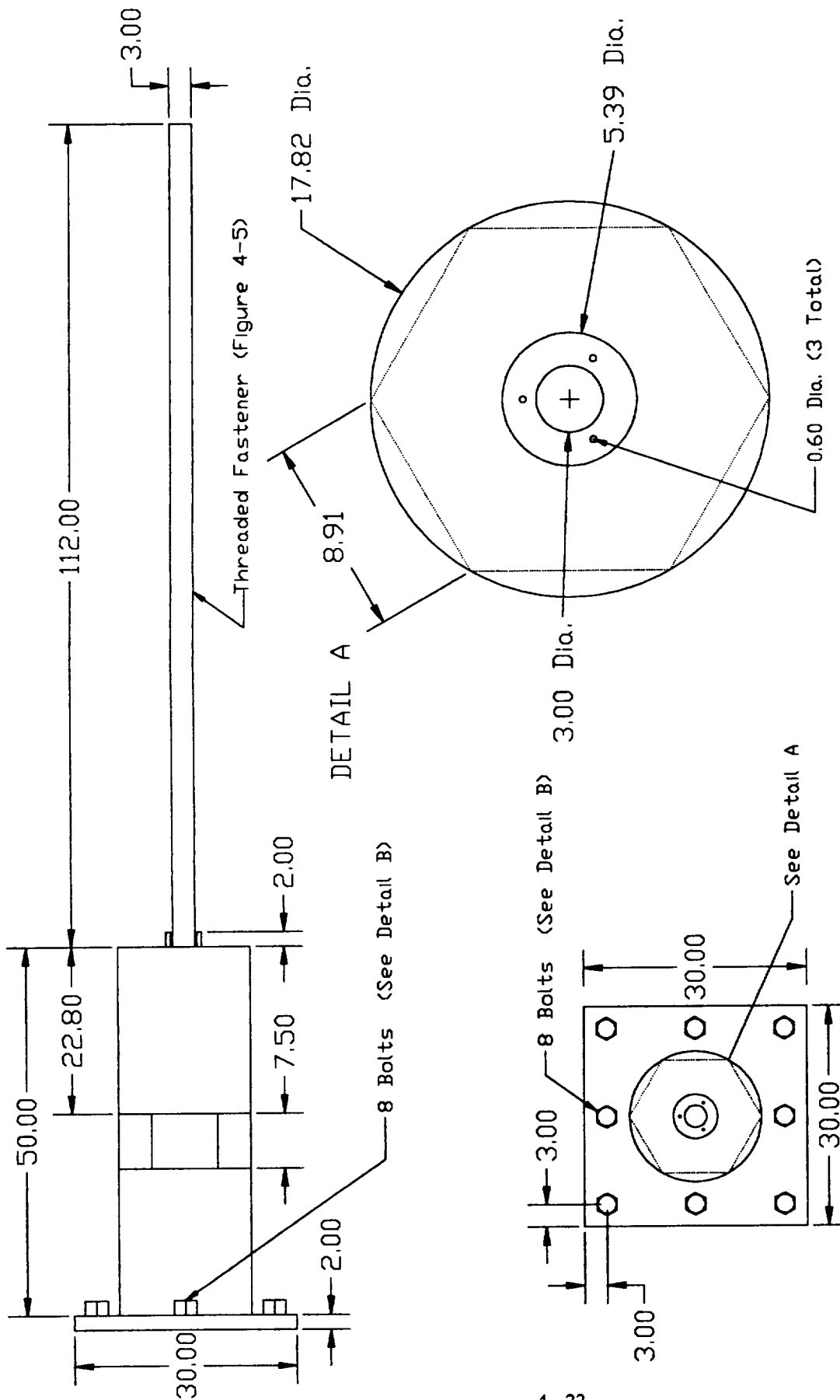
DETAIL A



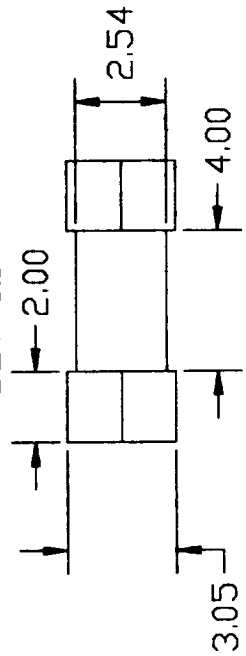
NOTES:

- 1. All dimensions in cm
- 2. Material is Aluminum

TITLE:	TROLLEY
DRAWN BY:	JLW
DATE:	3/10/94
ME	4182 MINING



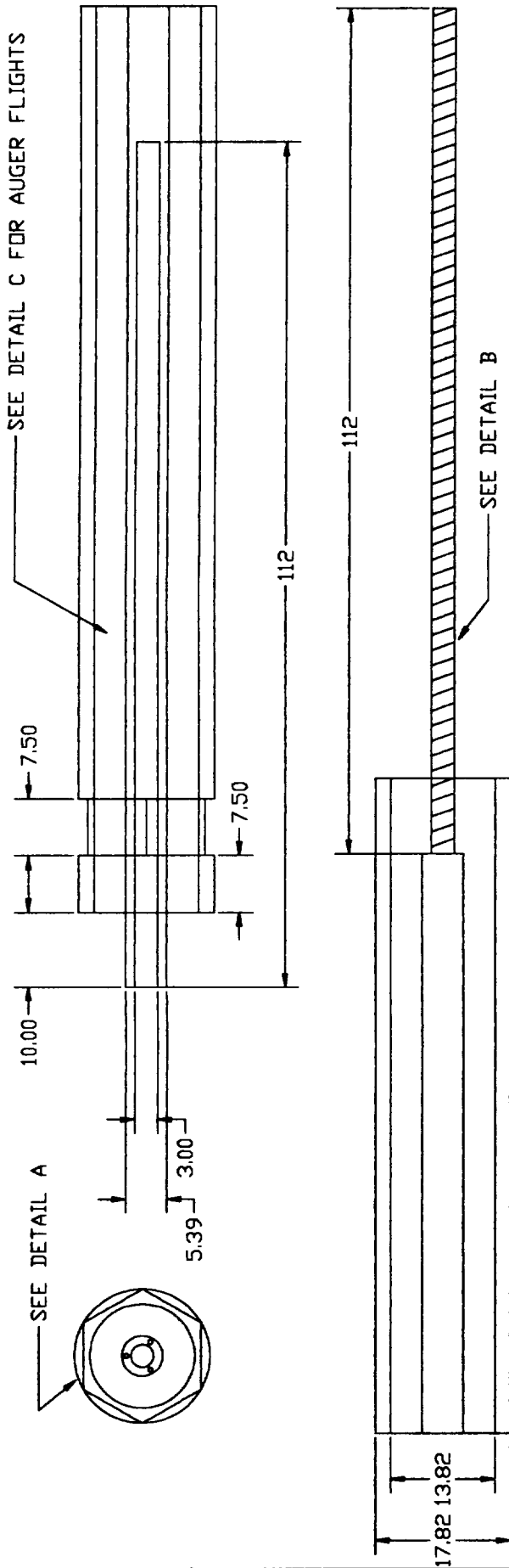
DETAIL B



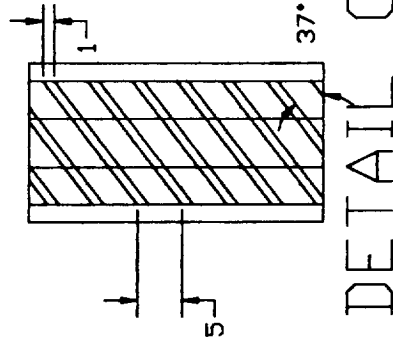
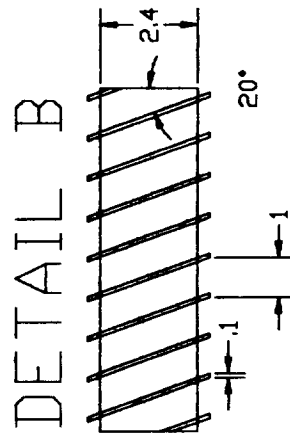
NOTES:

1. All dimensions are in cm.
2. Materials listed in text.
3. Bolts connect motor to trolley.

TITLE:	MOTOR
DRAWN BY:	ECB
DATE:	3/10/94
ME	4182 MINING

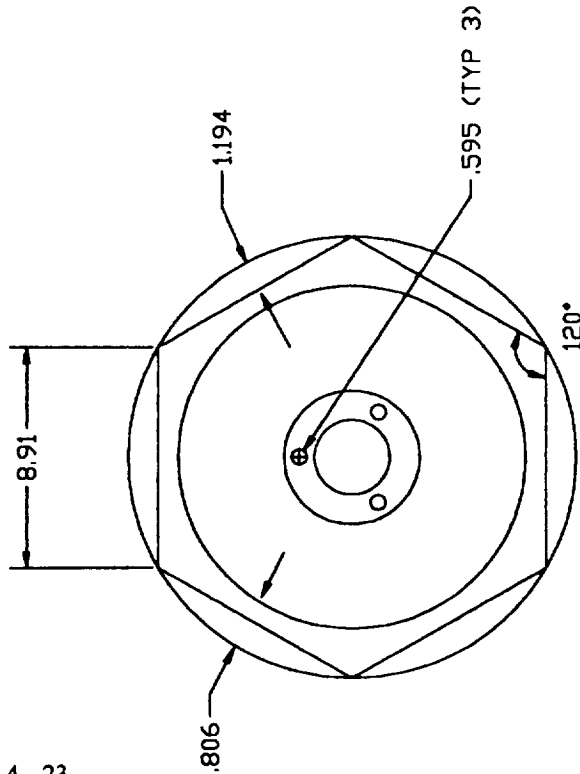


4 - 23

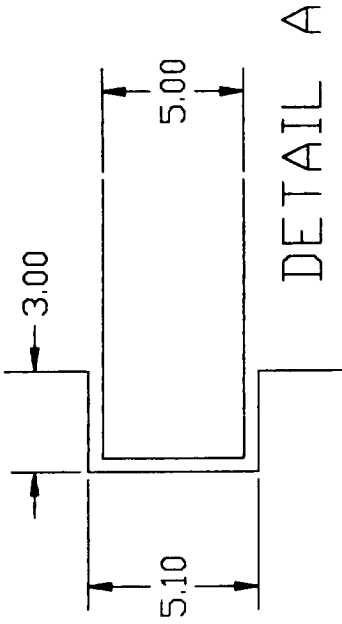
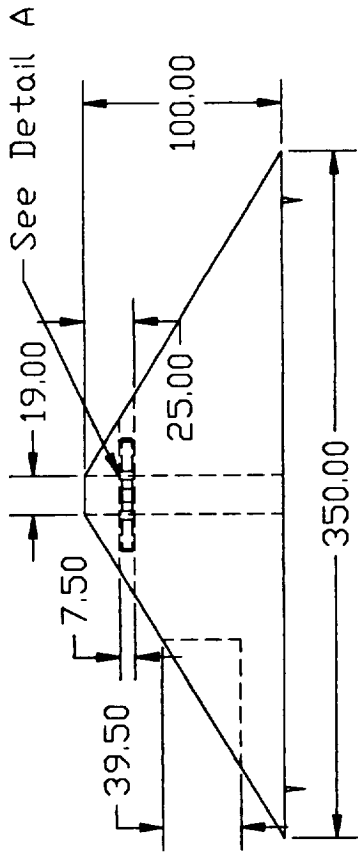


NOTES:

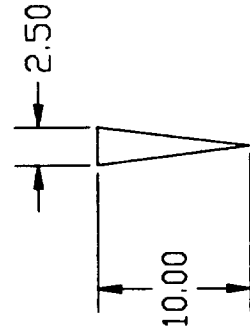
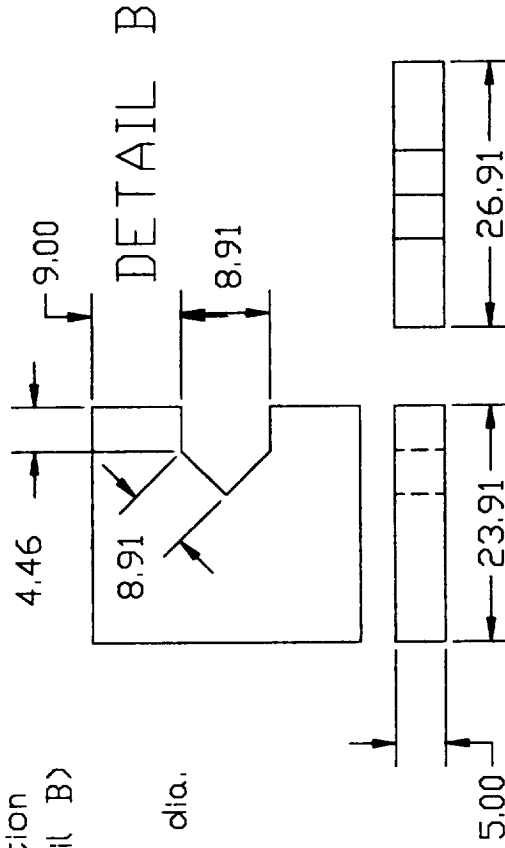
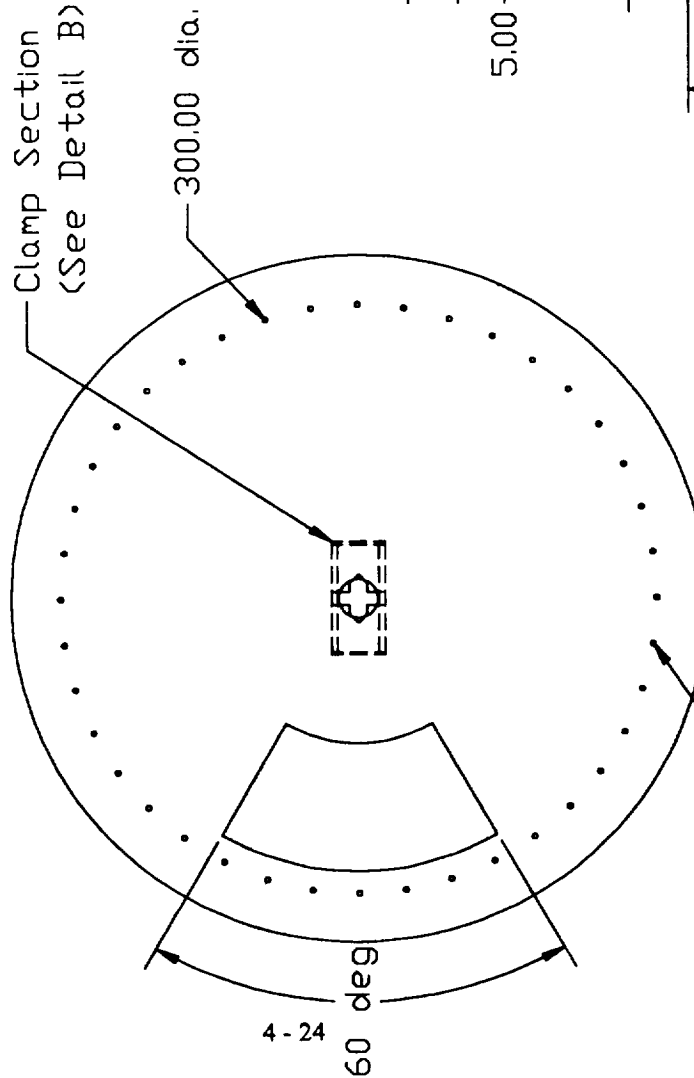
1. All dimensions in cm
2. Full length of auger sections is 16.2m



TITLE:	AUGER
DRAWN BY:	JLW
DATE:	3/10/94
ME	4182 MINING



Clamp Section
(See Detail B)



TITLE:	GRIPPER
DRAWN BY:	ECB
DATE:	3/10/94
ME	4182 MINING

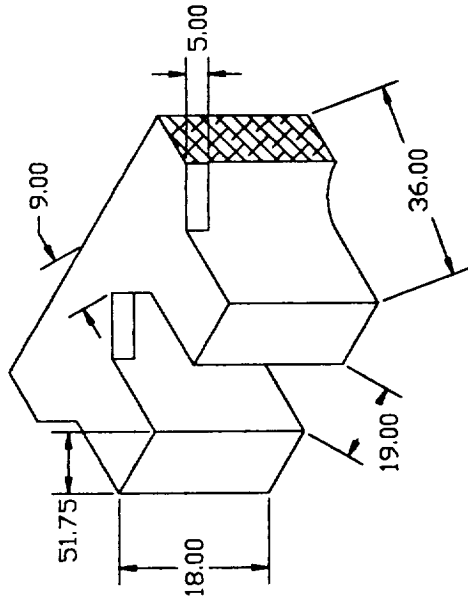
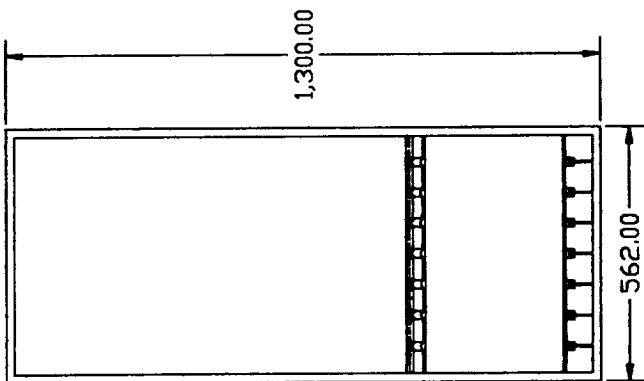
NOTES:

1. All dimensions are in cm.
2. Materials listed in text.
3. Clamp run by solenoid.
(Wiring not Shown)

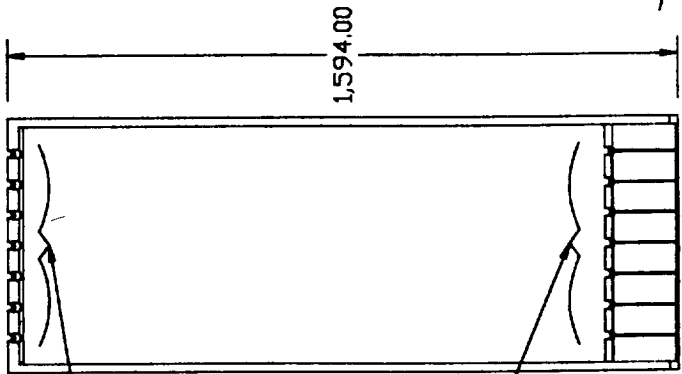
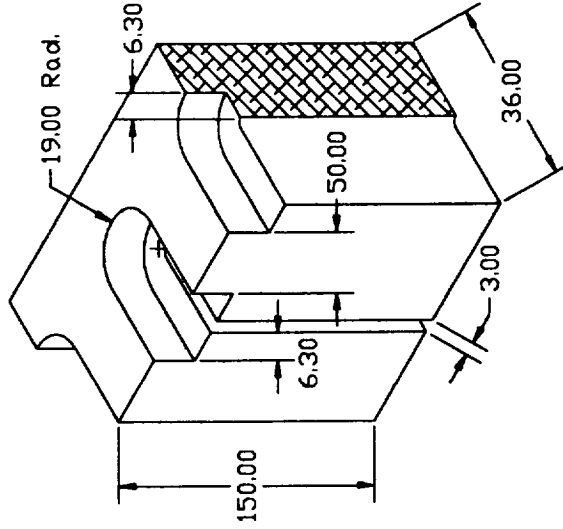
NOTES:

1. All dimensions are in cm.
2. Materials listed in text.

DETAIL A



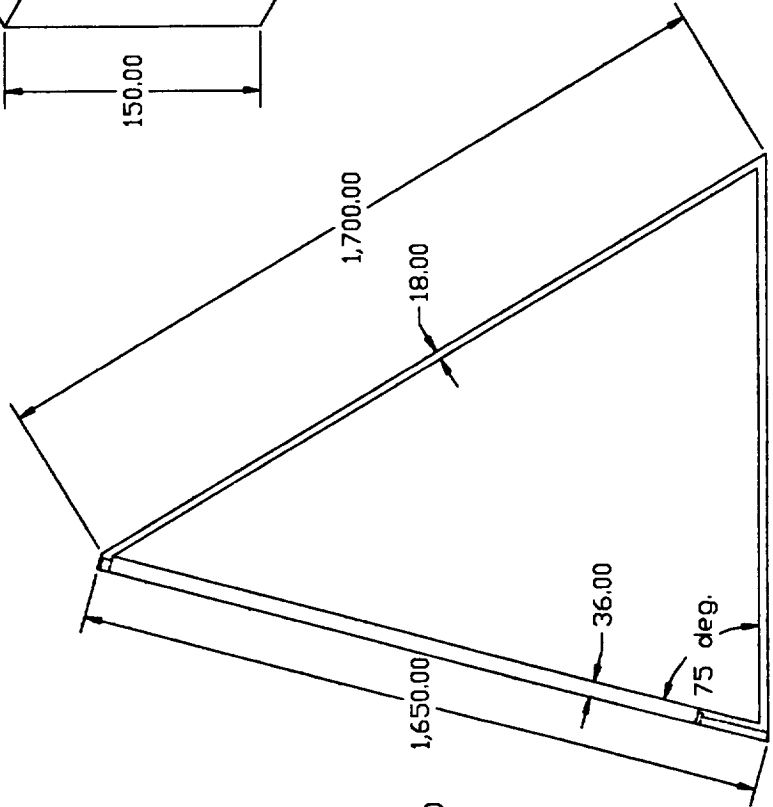
DETAIL B



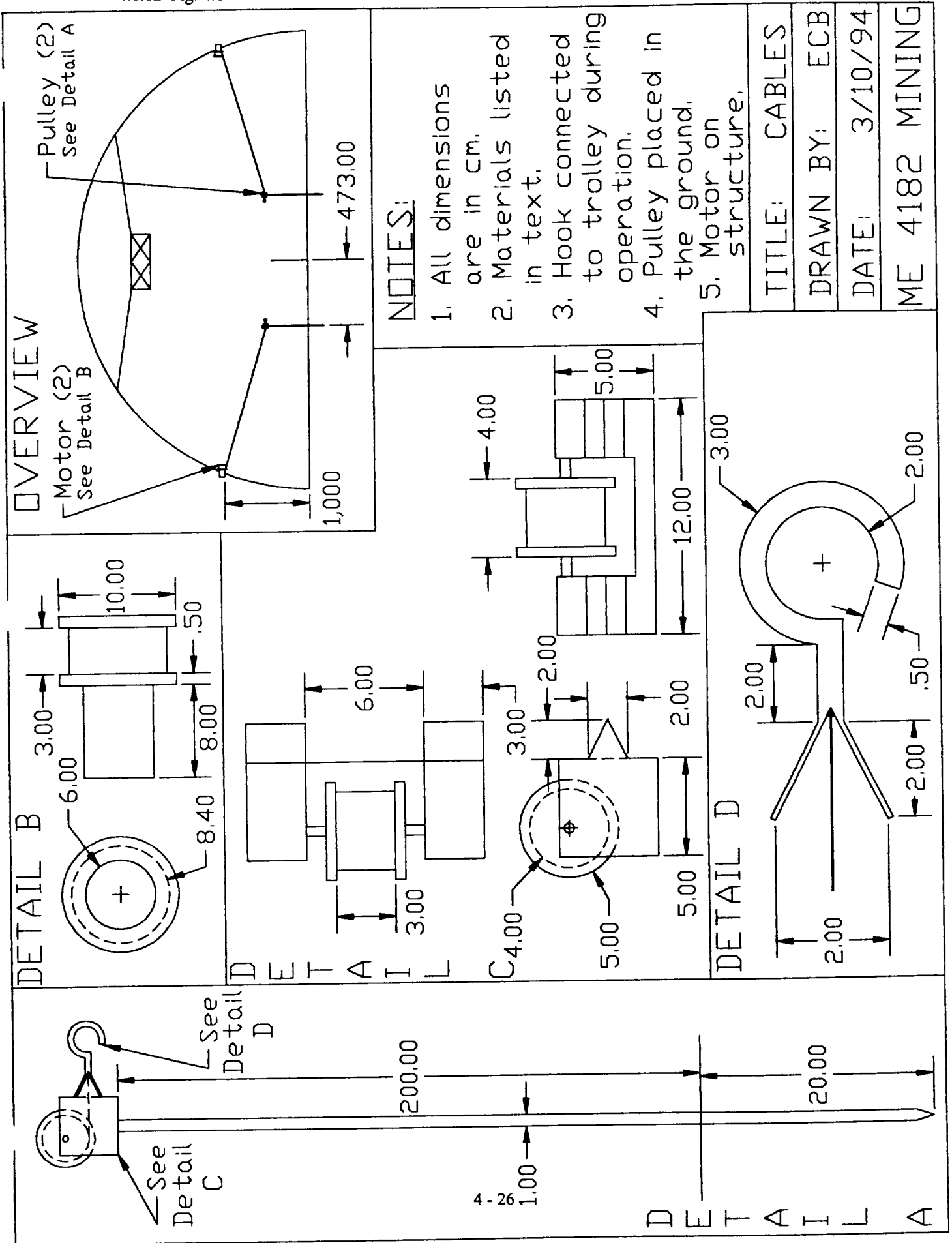
4 - 20

See Detail A

See Detail B



TITLE:	RACK
DRAWN BY:	ECB
DATE:	3/10/94
ME	4182 MINING



5 DOME CONSTRUCTION

Jaesuk Yoo
Theodore McCrobie

for

Professor Brazell
ME 4182
Winter '94

5.0 EXECUTIVE SUMMARY

The construction of a 45 meter tall and 30 meter diameter lunar mining dome by a dual armed, semi-autonomous robot will require support material (May Poles, wenchers, wires, etc.), the robot (including the arms and hands), and a construction process (a step-by-step method of construction).

To fulfill these requirements a 47.5 meter tall, 1 meter diameter, 1 millimeter thick May Pole has been designed with all the support material needed. This includes the wenchers needed to lift the dome, the supports during construction, and the jacks for assisting the lifting of the May Pole. See appendix 5.6.A for more information on the May Pole.

The robot end effector has been designed to hold the poles and the guide wire, as well as manipulate the poles and insert pins and spikes. The "hand" will be a three "fingered" hand with a complex wrist. The end effector chosen is the Stanford/JPL Hand mounted on a Powered Wrist Joint designed by Spar Aerospace Product Ltd. Also, the choice between a dual or single armed robot has been made. A dual armed *Enabler* will be needed to construct the dome adequately. Cameras will be mounted on the upper arm of the robot so as to better see the work being done by the *Enabler*. See appendix 5.6.B for detailed drawings of the hand and wrist

Also a detailed construction process has been written with estimated times of completion. The *Enabler* can build the dome structure in approximately 20 hours. See appendix 5.6.C for a more detailed list of the construction process.

5.1 PROBLEM STATEMENT

The Lunar Mining Dome designed must be built on the surface of the Moon. Within this task there are several areas for development. The "hand" of *The Enabler* must be designed for grasping and manipulating the construction pieces. Also, the may pole which will raise the dome as it is being built must be designed. Finally, the actual construction process must be designed. This consists of the order in which parts will be placed, any robotic movements, and the time associated with all construction actions.

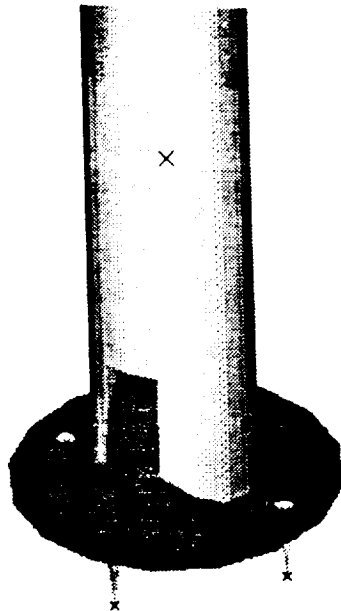
The design is limited by various factors. The construction equipment must be transportable to the construction site. This means the may pole, robot, and any support materials must first transported to the Moon by a Shuttle and other space vehicle, then transported to the construction site on the lunar surface. Weight and size are the two major factors. At the construction site there will be only one robot. Therefore, the dome construction process must be performed by one single or dual armed robot, *The Enabler*. Also, this robot must be mostly autonomous. And, all of the construction should be completed as quickly as possible.

Once the construction process is completed, the robot, may pole, and any other construction tools must be reused at another site. This factor forces any tools used at the construction site to be easily transportable as well as durable.

5.2 May Pole Proposal

The May Pole is used to support the dome structure while being built. The dome will be built in a top down method. To build the dome this way without lifting the robot, the dome must be lifted. The May Pole will lift the dome with the use of powered wenches as the structure is built.

Since the dome is 45 meters tall the May Pole must be taller to accommodate the wench and guy wire system designed. The May Pole needed to be 47.5 meters in length. However, a 47.5 meter pole would be extremely hard to manipulate so it was divided into sections.



The May Pole will consist of three 12.5 meter poles with an overlap of 2.5 meters and one 15 meter pole. The top pole will have the wench system fitted to it, and the bottom pole will be designed to fit into a pinned lifting system.

To keep the structure light, a carbon fiber matrix with a strong epoxy resin will be used. The carbon fiber utilized for this structure is IM-

6 material produced by the Hercules Corporation. The resin will be resin 3501-6 cured at 350 °C. These materials were chosen for their strength and light-weight. The epoxy resins are excellent adhesives which bond easily to most fibers. The resin forms a strong interface, so laminates made with the resin show a large degree of the fiber properties. The carbon fiber was chosen because of its great balance of properties such as high compression, fatigue, and impact strengths.

The dome structure is estimated to have a mass of approximately 1000 kg. With the maximum stress in compression for the material being 1.724 GPa and utilizing a safety factor of 4, the optimum diameter and thickness for the May Pole was found to be 1.0 meters and 1 millimeter, respectively. Those dimensions were derived from the following equations:

$$\sigma_{\max}=P/A \quad (5.1)$$

where, $A=\pi t(d-t) \quad (5.2)$

where, P is the force exerted by the dome on the May Pole, A is the cross-sectional area, d is the diameter, and t is the thickness of the shell.

After determining the dimensions of the May Pole which would support the dome, buckling had to be taken into account. Using:

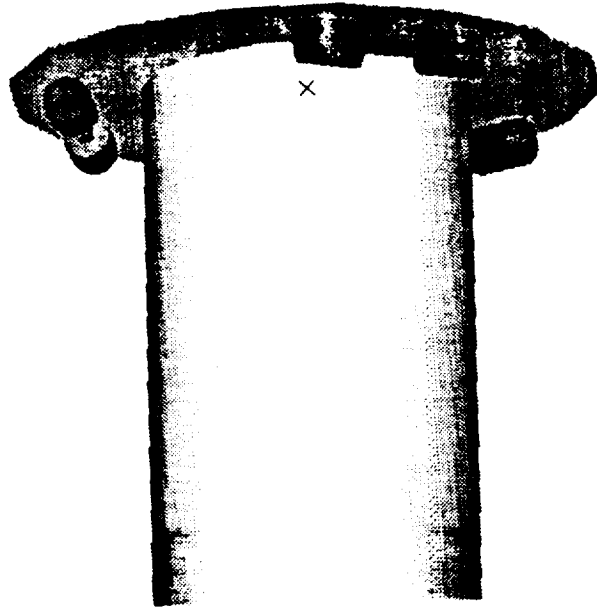
$$P_{cr}=\pi^2 EI/4L^2 \quad (5.3)$$

where, $I=(\pi/2)(r_o^4-r_i^4) \quad (5.4)$

where, E is Young's modulus, L is the total length, r_o is the outer radius, and r_i is the inner radius, the dimensions chosen were found to resist buckling. Just to enhance the safety of the May Pole, a light-weight foam was used to fill the hollow of the May Pole.

The top cap of the May Pole will hold the motors which will lift the dome. The disk is made of Aluminum Alloy 2024 in a honeycomb

structure. This was chosen due to its strength and low density. When stress analysis was performed, the disk was found to be 1.2 cm thick.



The motors used to lift the dome will be Bodine Type Number 34B3•EBL-E4. This single shafted 1/5 Hp motor is capable of delivering 350 Lb-in torque. To lift the dome, two of these motors will be connected to a 20 cm drum to reel in the cables. Three of these pairs will be used on the May Pole top.

Kevlar cables will be used to lift the dome. Kevlar is very light weight, .130 g/cm³, and very strong, 7 GPa in tension. A 5 mm cable was chosen, even though it far exceeds the minimum diameter, so that the *Enabler* can better handle this cable.

The pinned base used to assist the lifting of the May Pole will also be made of Aluminum Alloy 2024 in a honeycomb structure. The disk will be 2 m in diameter with a 1 meter cylinder in the center. The base will have 4 holes in its base for the screw spikes. A 10 cm rod made of Aluminum Alloy 6063-T6 will be used as the pivot.

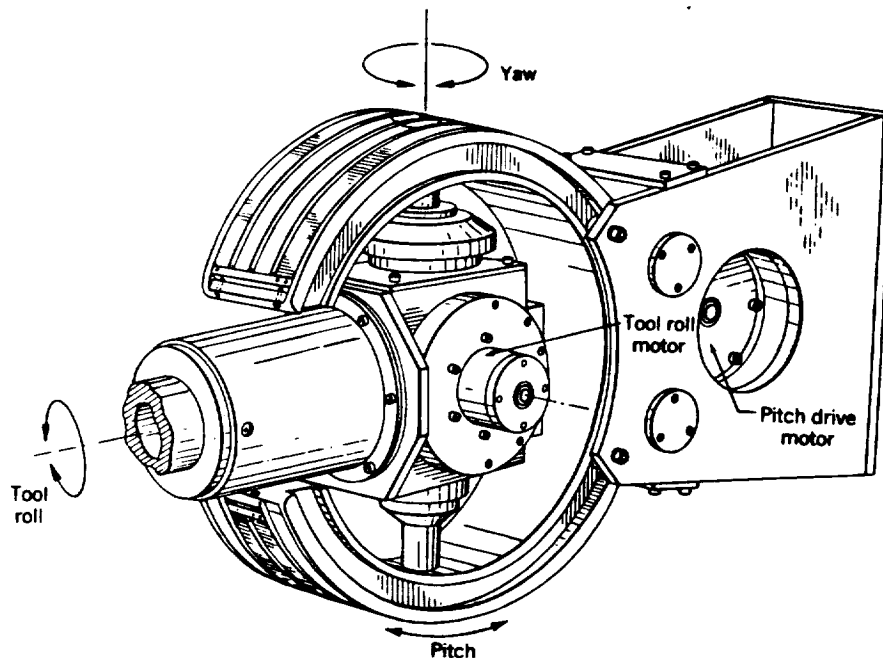
5.3 Robot End effector Proposal

The end effector of the Enabler must be designed so that it can perform all of the functions needed by the hand during the construction of the dome. The tasks to be performed by the *Enabler* are:

- lift and manipulate the bars of each level
- lock the rings of each level together
- twist the bars in to the locks on the rings
- grasp and maneuver the supply carts
- grasp and manipulate the May Pole pieces
- grasp any support systems designed
- grasp and manipulate the cables for lifting

To achieve these a three "fingered" end effector was designed with a complex wrist. The hand is the Stanford/JPL Hand patent # XXXXX. (Picture in Appendix 5.6.B-1) This hand is capable of grasping the 5 cm pole and the .5 cm cable.

The wrist is the Powered Wrist Joint designed by Spar Aerospace Productions Ltd. patent # XXXXXX. The wrist has a pitch capacity of 270 degrees and a yaw capacity of 180 degrees. It will also allow the Enabler to roll the hand a continuous 360 degrees.



The pitch and yaw will allow the *Enabler* to have fine adjustment control when inserting poles into locks and will give the hand the ability to twist the pole to complete the lock. The continuous roll will allow the *Enabler* to screw in the spikes used to secure the May Pole base and the wenchers. The fingers will be coated with a rubber material to create the friction needed to generate torque on the poles.

The motors that control the wrist are located in the wrist. This eliminates the need for hydraulic or tension wires to control it. The hand however uses tension cables to power the fingers. This will result in the finger motors needing to be attached between the wrist end and the hand base.

It is recommended that the *Enabler* be fitted with a second arm.

This is recommended for three reasons:

1. The complexity of the locking mechanisms requires that one hand holds the lock while the other secures the pole into it.
2. The second arm will reduce the shear stress placed on the poles as it is manipulated into position.
3. The second arm will greatly reduce the extra motions of the poles while being manipulated, thus making them easier to wield.

5.4 Construction Process Evaluation

The process of building the dome can be divided into several sub-processes. These include the preparation of the construction site, construction of the May Pole, the dome construction, and the clearing of the construction site. The following is an outline of the construction process:

- I. Preparation
 1. Positioning of wenches at 80 meters distance from May Pole
 2. Placement of May Pole base at construction site
- II. May Pole Construction
 1. Obtaining May Pole parts
 2. Placement of supports
 3. Placement of pinned bottom pole
 4. Insertion of 4 other poles
 5. Placement of Top ring over May Pole
 6. Insertion of lifting wenches top pole
 7. Connection of all cables
 8. Placement of jacks for lifting
 9. Lift pole to vertical
 10. Lock down guy wires
- III. Dome Construction
 1. Obtain levels poles and ring
 2. Construct ring
 3. Add all poles for level
 4. Mount cameras
 5. Lift dome
 6. Repeat 1-5 for all nine levels
 7. Mount Stewart Platform cables (on fourth level)
- IV. Finishing
 1. Set dome on Lunar surface
 2. Disassemble May Pole
 3. Remove to staging area
 4. Gather wenches

The estimates of the time needed to complete the construction, step-by-step are located in Appendix C. From that spreadsheet the adjusted estimated time of construction is 20 hours.

5.5 Building Supplies and Features

During the construction of the May Pole and dome several aids will be used to make the construction easier. These include:

- Wenches to lift the dome
- Lifting Jack to assist the wenches in lifting the May Pole
- Supply carts to move materials more easily
- Support "saw-horses" during the May Pole construction
- G Clamps on cables for easy attachment to May Pole

The descriptions for each of these can be found in appendix 5.6.D.

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Scott Warham of Hercules Aerospace.

Jonathan Colton, Georgia Institute of Technology, Mechanical Engineering Department, MRC 434.

Wayne J. Book, Georgia Institute of Technology, Mechanical Engineering Department, MRC 472.

"Brushless DC Motors Catalog", Bodine Electric Company, 1993.

Appendix Summary

Analysis of May Pole	A-1
Analysis of May Pole Base	A-2
Analysis of Pin	A-3
Analysis of May Pole Cap	A-4
Analysis of Kevlar Cable	A-5
Stress and Buckling Analysis of May Pole	A-6
CAD Drawings	A-7
Staford/JPL Hand	B-1
Powered Wrist Joint	B-2
Time of Construction Analysis	C-1
Wench Description	D-1
Jack Description	D-2
Cart Description	D-3
Support Description	D-4

CARBON IM-6 3501-6 EPOXY RESIN

$$\sigma_u = 1,723,690,000 \text{ Pa} ; P_{\text{BONE}} = 1000 \text{ kg} = 1635 \text{ N}$$

$$E = 131,000,440,000 \text{ Pa} ; P_{\text{CAP}} = 80 \text{ N}$$

$$\text{SAFETY FACTOR } (n) = 4 ; P = 1715 \text{ N} ; \text{SHEAR STRENGTH} = 68.95 \text{ MPa}$$

$$\sigma_{\text{ALLOW}} = \frac{\sigma_u}{n} = 1,436,408,333 \text{ Pa}$$

$$A_{\text{CROSS-SEC}} = \frac{P}{\sigma_{\text{ALLOW}}} = 1.194 \times 10^{-6} \text{ m}^2$$

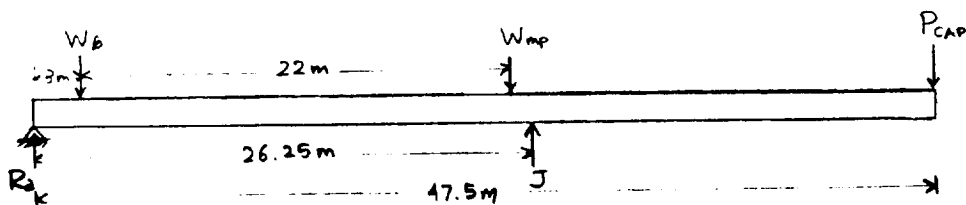
FOR BUCKLING CONSIDERATION, THE CRITICAL LOAD IS

$$P_{\text{CR}} = \frac{\pi^2 EI}{4L^2}$$

$$\text{WHERE } I = \frac{\pi}{2} (r_o^4 - r_i^4)$$

$$r_o = 0.5 \text{ m}$$

* RESULTS ARE ON PAGE 56. A-6



$$\sum M_B = -3 W_B - 25 W_{mp} + 26.25 J - 47.5 P_c = 0$$

$$\text{WHERE WITH } r_i = 0.499 \text{ m } A_{\text{CROSS-SEC}} = \pi(r_o^2 - r_i^2) = 3.138 \times 10^{-3} \text{ m}^2$$

$$L_{mp} = 45 \text{ m} + 10 \text{ m (OVERCAPS)} = 55 \text{ m}$$

$$V = AL = 1.726 \times 10^{-1} \text{ m}^3$$

$$\text{DENSITY}_{mp} = 1590 \text{ kg/m}^3$$

$$W_{mp} = (274.458 \text{ kg})(1.635 \text{ m/s}^2) = 448,739 \text{ N}$$

ALUMINUM ALLOY 7075-T6

$$\sigma_u = 550 \text{ MPa}$$

$$P = P_{\text{DOME}} + P_{\text{CAP}} + P_{\text{MP}} = 2164 \text{ N}$$

$$E = 72 \text{ GPa}$$

$$\text{SAFETY FACTOR } (n) = 4$$

$$\sigma_{\text{ALLOW}} = \frac{\sigma_u}{n} = 137.5 \text{ MPa}$$

$$A_{\text{CROSS-SEC}} = \frac{P}{\sigma_{\text{ALLOW}}} = 1.574 \times 10^{-5} \text{ m}^2 < 3.138 \times 10^{-5} \text{ m}^2 \quad (\text{FOR } r_o = 0.5 \quad r_i = 0.499)$$

FOR BUCKLING CONSIDERATION,

$$P_{\text{CR}} = \frac{\pi^2 EI}{4L^2}$$

$$\text{WHERE } L = 0.5 \text{ m}$$

$$P_{\text{CR}} = 556 \text{ MN} > P \quad (\text{OKAY})$$

$$\text{DENSITY}_{\text{AL}} = 2800 \text{ kg/m}^3$$

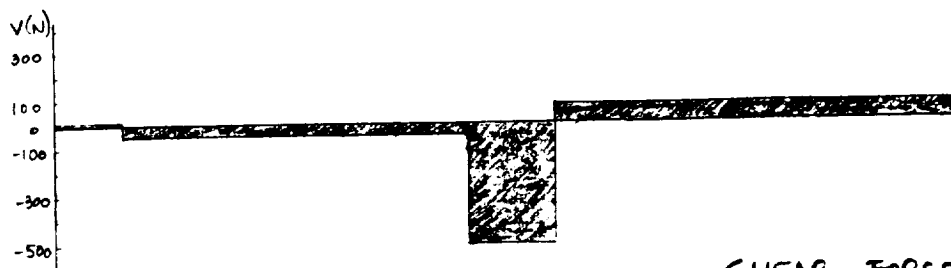
$$W_B = (35.146 \text{ kg})(1.635 \text{ m/s}^2) = 57.463 \text{ N}$$

$$\sum M_2 = -3(57.463 \text{ N}) - 25(448.739 \text{ N}) + 26.25 \text{ J} - 47.5(80 \text{ N}) = 0$$

$$J = 578.700 \text{ N} \quad (\text{FORCE FROM JACK})$$

$$\sum F_y = R_2 - W_B - W_{\text{MP}} + J - P_c = 0$$

$$R_2 = 7,503 \text{ N}$$



MAX = -500 N

SHEAR FORCE DIAGRAM

FOR HOLLOW CIRCULAR CROSS SECTION OF COMPOSITE

$$\tau_{max} = \frac{VQ}{Ib}$$

WHERE $I = \frac{\pi}{4} (r_o^4 - r_i^4)$

$$b = 2(r_o - r_i)$$

$$Q = \frac{2}{3} (r_o^3 - r_i^3)$$

$$\tau_{max} = \frac{4V}{3A} \frac{r_o^2 + r_o r_i + r_i^2}{r_o^2 + r_i^2}$$

WHERE $A = \pi (r_o^2 - r_i^2)$

$$V = 500 \text{ N (FROM SHEAR FORCE DIAGRAM)}$$

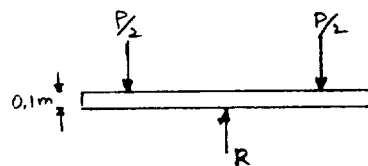
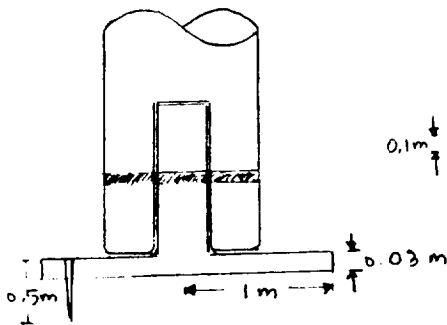
SUBSTITUTING...

$$\tau_{max} = 318628 \text{ Pa} \quad (\text{IM-6 3501-6})$$

THIS IS LESS THAN MAX SHEAR STRENGTH OF MATERIAL

WHICH IS 68.95 MPa SO OKAY.

ALUMINUM 6063-T6 BASE PIN



$$P = P_{\text{DOME}} + P_{\text{CAP}} + W_{\text{NP}} + W_{\text{BASE RUL}} \\ = 1635 + 80 + 448.739 + 57.463$$

$$P = 2221.202 \text{ N}$$

$$\sum F_y = R - P/2 - P/2 = 0$$

$$R = P = 2221.202 \text{ N} = V$$

$$\tau_{max} = \frac{VQ}{Ib}$$

WHERE $I = \frac{\pi r^4}{4}$

$$b = 2r$$

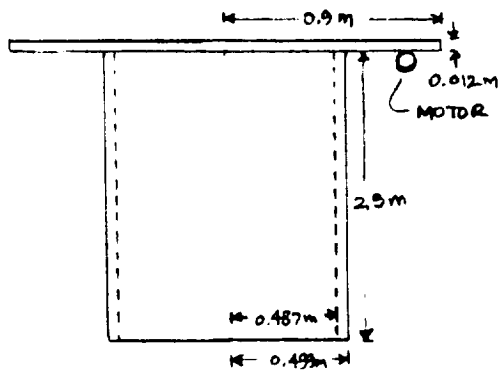
$$Q = \frac{\pi r^2}{2} \left(\frac{4r}{3} \right) = \frac{2r^3}{3}$$

$$\tau_{max} = \frac{4V}{3A} = \frac{4V}{3\pi r^2} = 377082.963 \text{ Pa}$$

THIS IS LESS THAN MAX SHEAR STRENGTH OF

AL 6063-T6 WHICH IS 250 MPa SO OKAY.

ALUMINUM ALLOY 2024 HONEYCOMB STRUCTURE CAP



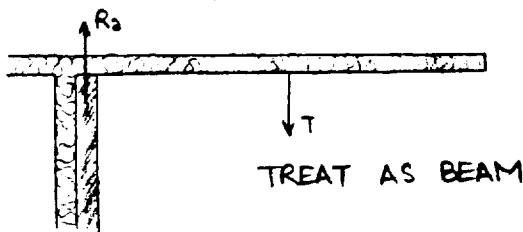
$$\text{DENSITY} = 0.130 \text{ g/cm}^3 = 130 \text{ kg/m}^3$$

$$\text{SHEAR STRENGTH} : 7 \text{ MPa}$$

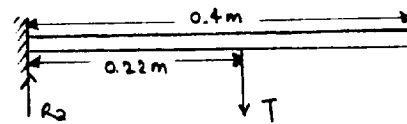
$$\text{VOL} = \pi (0.9 \text{ m})^2 (0.012 \text{ m}) + \pi [(0.499)^2 - (0.487)^2] (2.5 \text{ m})$$

$$= 1.235 \times 10^{-1} \text{ m}^3$$

$$W_{\text{cap}} = (16.050 \text{ kg})(9.81 \text{ m/s}^2)(1/6) = 26.242 \text{ N}$$



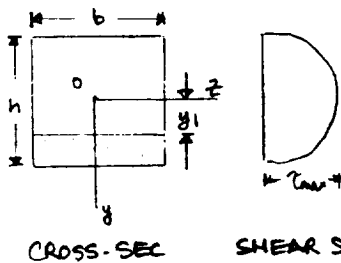
TREAT AS BEAM



$$T = W_{\text{DOME}} + W_{\text{MOTOR \& BEAM}} + W_{\text{CABLE}}$$

$$T = 545 + 17 + 2 = 564 \text{ N} = R_2 = V$$

* WEIGHT OF BEAM NEGLIGIBLE



CROSS-SEC

SHEAR STRESS

- MAX SHEAR STRESS OCCURS AT
THE NEUTRAL AXIS, $y_1 = 0$.

$$\tau_{\text{max}} = \frac{3V}{2A} = \frac{3V}{2bh}$$

$$\text{WHERE } b = h = 0.012 \text{ m}$$

$$\tau_{\text{max}} = 5.875 \text{ MPa} < 7 \text{ MPa} \text{ SO OKAY.}$$

- THIS IS τ_{max} FOR A BEAM. THE
 τ_{max} FOR THE DISK WILL BE
SIGNIFICANTLY SMALLER.

KEVLAR CABLES

$$\text{WEIGHT OF DOME} = (1000 \text{ kg})(9.81 \text{ m/s}^2)(\frac{1}{6}) = 1635 \text{ N}$$

6 MOTORS SO EACH MUST SUPPORT 272.5 N

$$1.44 \text{ g/cm}^3 = 1440 \text{ kg/m}^3 \quad (\text{DENSITY}) \quad (2 \text{ MOTORS TOGETHER})$$

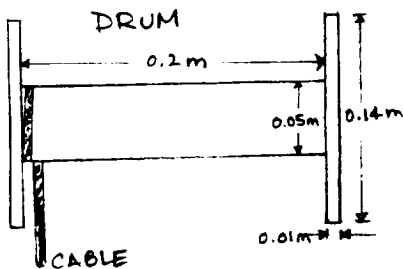
$$\sigma_u = 7 \text{ GPa}$$

$$\text{SAFETY FACTOR } (n) = 4$$

$$\sigma_{\text{ALLOW}} = \frac{\sigma_u}{n} = 1.75 \text{ GPa}$$

$$\text{ACROSS-SEC} = \frac{P}{\sigma_{\text{ALLOW}}} = \frac{545 \text{ N}}{1.75 \text{ GPa}} = 3.114 \times 10^{-7} \text{ m}^2$$

- CABLE DIAMETER OF 0.005 m WILL BE USED SO THAT THE ENABLER WILL BE ABLE TO GRASP.
 - THIS GIVES A CROSS-SECTIONAL AREA OF $1.963 \times 10^{-5} \text{ m}^2$
 - EACH SET OF MOTORS WILL BE UTILIZING 50 METERS OF CABLE, SO TOTAL CABLE NEEDED IS 150 m
- $$V = AL = 2.945 \times 10^{-3} \text{ m}^3$$
- $$\text{CABLE WEIGHT} = (V)(\text{DENSITY})(9.81 \text{ m/s}^2)(\frac{1}{6}) = 6.933 \text{ N}$$



BECAUSE DIAMETER OF CABLE IS 0.5 cm, 40 ROTATIONS WILL OCCUR FOR EACH CABLE LAYER.

DIAM OF DRUM	LENTH OF CABLE WOUND FOR EACH LAYER
0.05	6.283
0.06	7.540
0.07	8.796
0.08	10.053
0.09	11.310
0.10	12.566

TOTAL 50.265 m

APPROXIMATELY 220 REVOLUTIONS ARE NECESSARY TO LIFT THE DOME TO 45 METERS.

STRESS AND BUCKLING ANALYSIS				
r (inner) (m)	Cross-section		Buckling	stress
	A (m ²)	P (critical) (N)	Thickness (m)	
0.499	0.003138	112179	0.001	519607.9
0.498	0.006271	223686	0.002	260064.3
0.497	0.009397	334523	0.003	173550.1
0.496	0.012516	444693	0.004	130293.3
0.495	0.015629	554199	0.005	104339.4
0.494	0.018736	663043	0.006	87036.9
0.493	0.021837	771228	0.007	74678.2
0.492	0.024932	878756	0.008	65409.3
0.491	0.028020	985631	0.009	58200.3
0.490	0.031102	1091855	0.010	52433.2
0.489	0.034177	1197431	0.011	47714.7
0.488	0.037247	1302361	0.012	43782.8
0.487	0.040310	1406648	0.013	40455.8
0.486	0.043367	1510294	0.014	37604.2
0.485	0.046417	1613302	0.015	35132.9

SDRC I-DEAS V6.1(s): Solid_Modeling

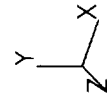
15-MAR-94 05:33:11

Unit: IN
Display: No stored Option

Database: MY_PDE
View: No stored View

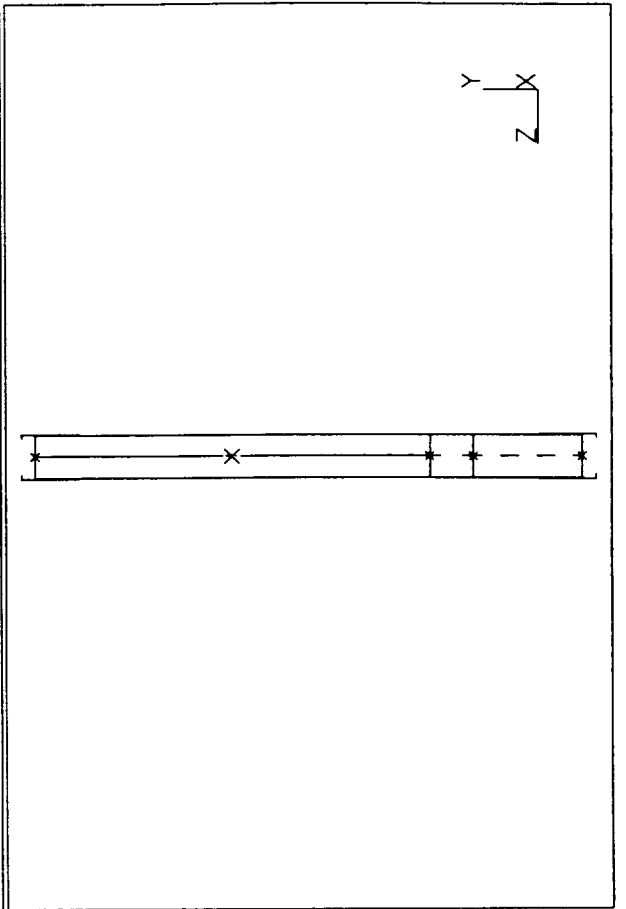
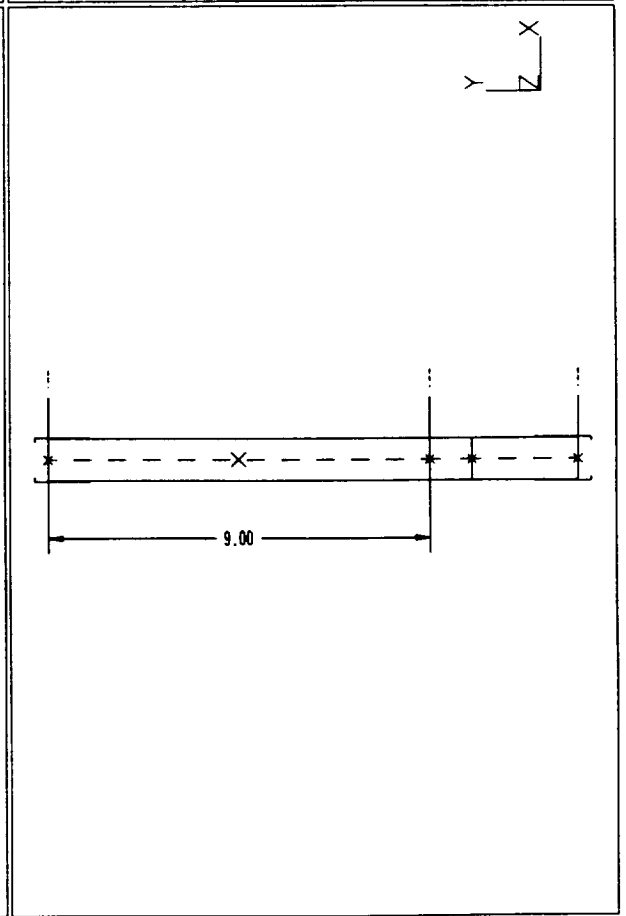
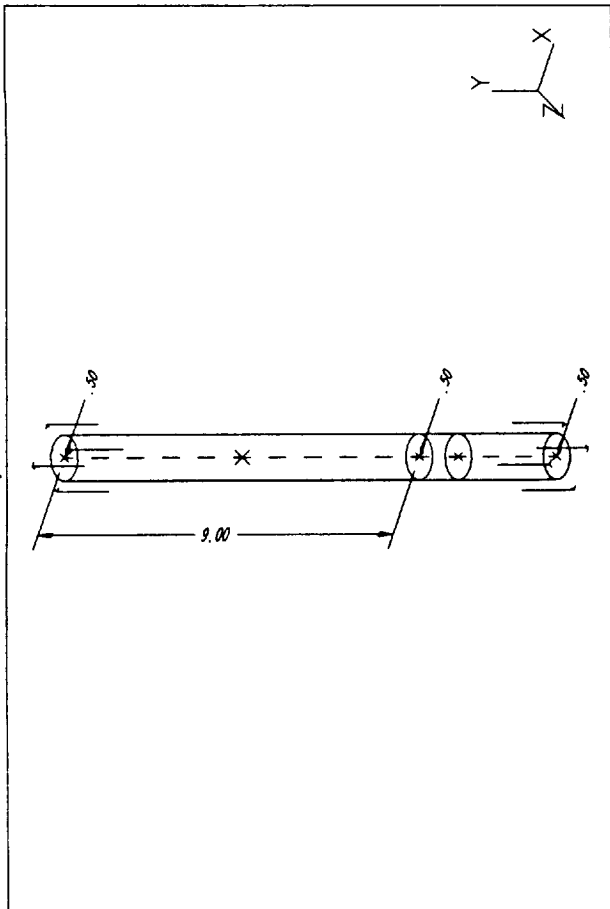
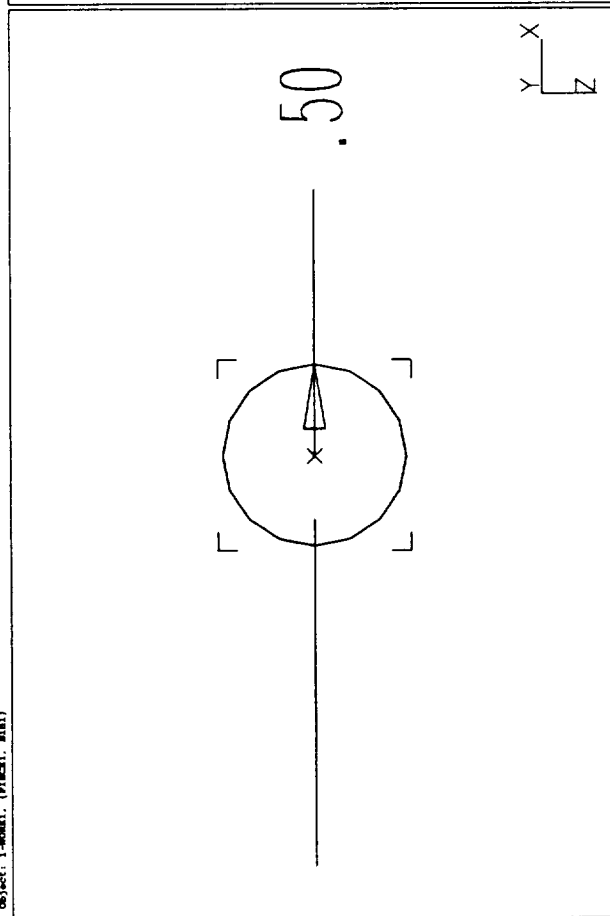
Task: Object
Object: 1-WORK1. (PIEC1, B1n1)

Bin: 1-WORK1
Object Level: Full



Database: MAT_PDL2
View : model, none, none, none
Task: Object
Object: 1-DEAS1, (PENCE, B1A1)

Display : none, none, none, none
Bin: 1-DEAS1
Update Level: Full



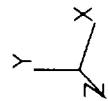
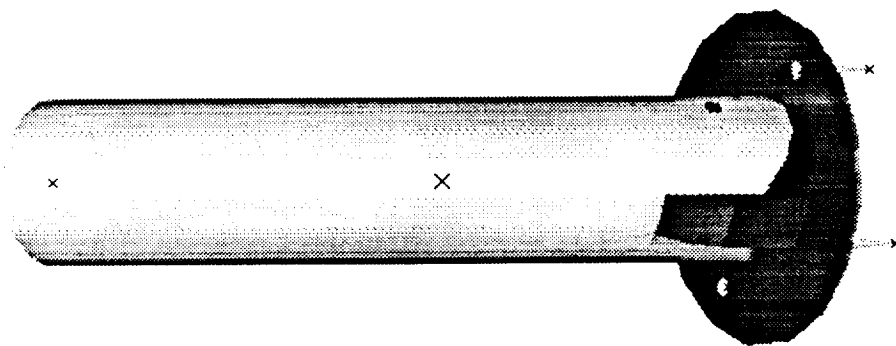
SDRC I-DEAS V6.1(6): Solid_Modeling

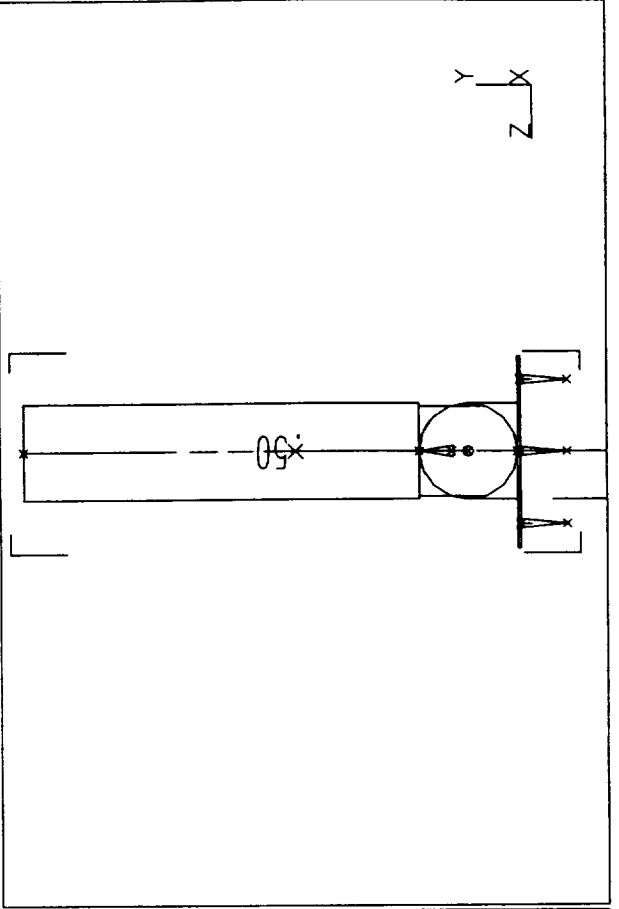
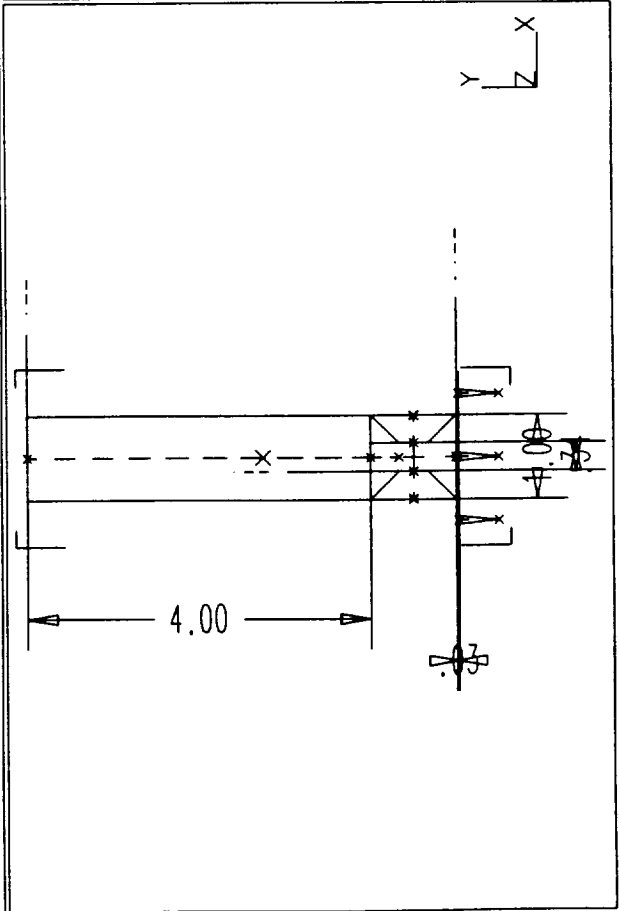
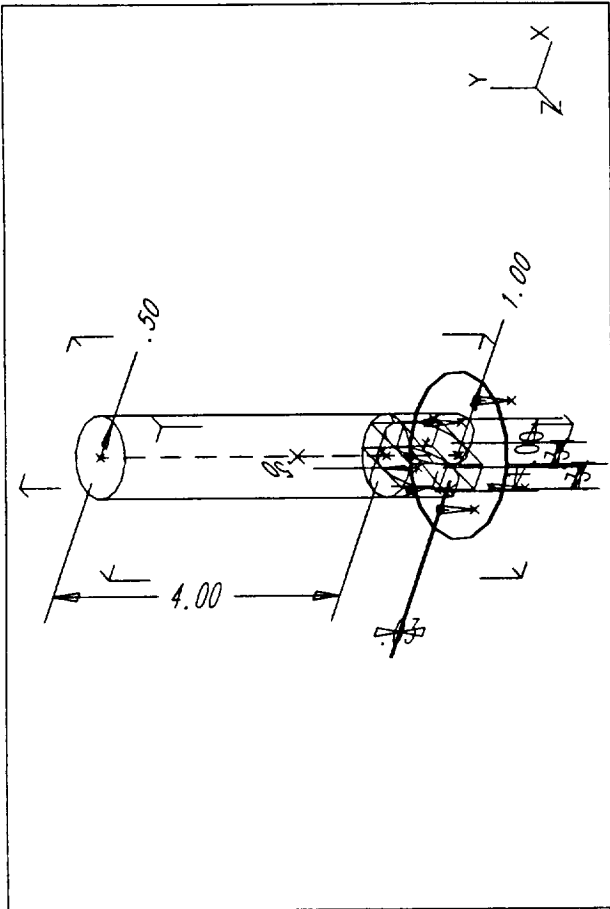
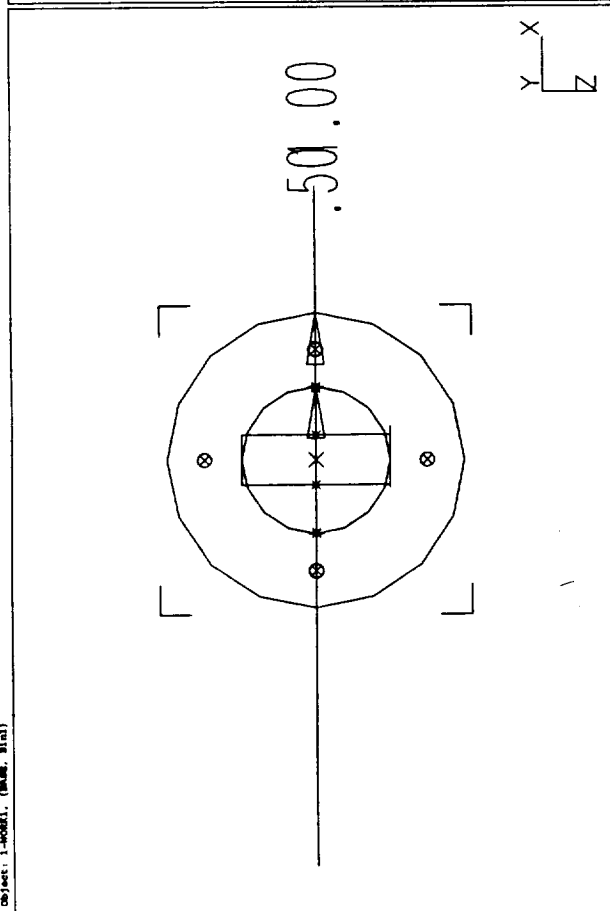
15-MAR-94 14:12:01

Units : IN
Display : No stored Option

Datbase: INT_POLE
View : No stored View
Task: Object
Object: I-MODEL (NAME, B1A1)

File: I-MODEL
Update Level: Pull

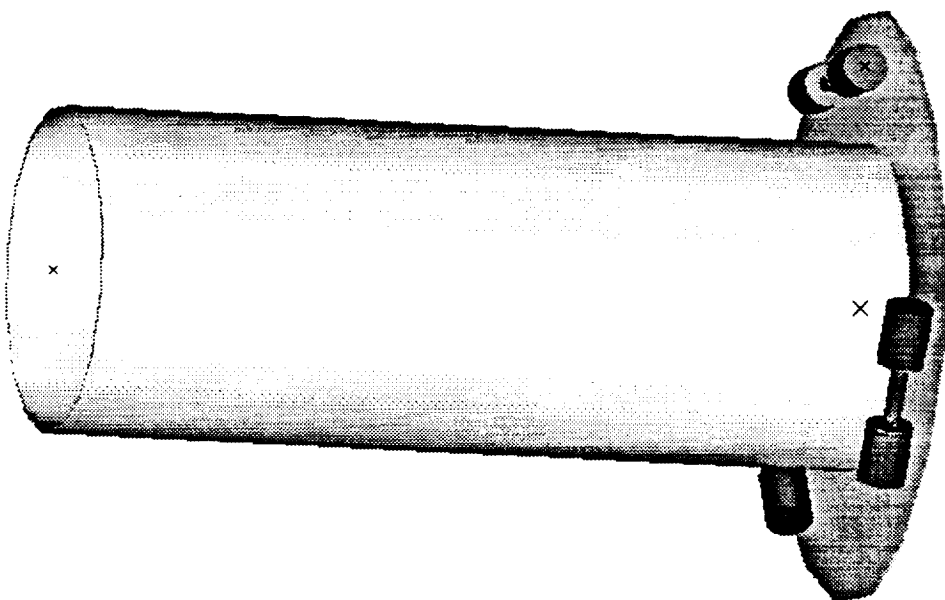




Database: MY_PCL
View : No stored View
Test Object
Object: 1-WIRE1, (CNR, SL1)

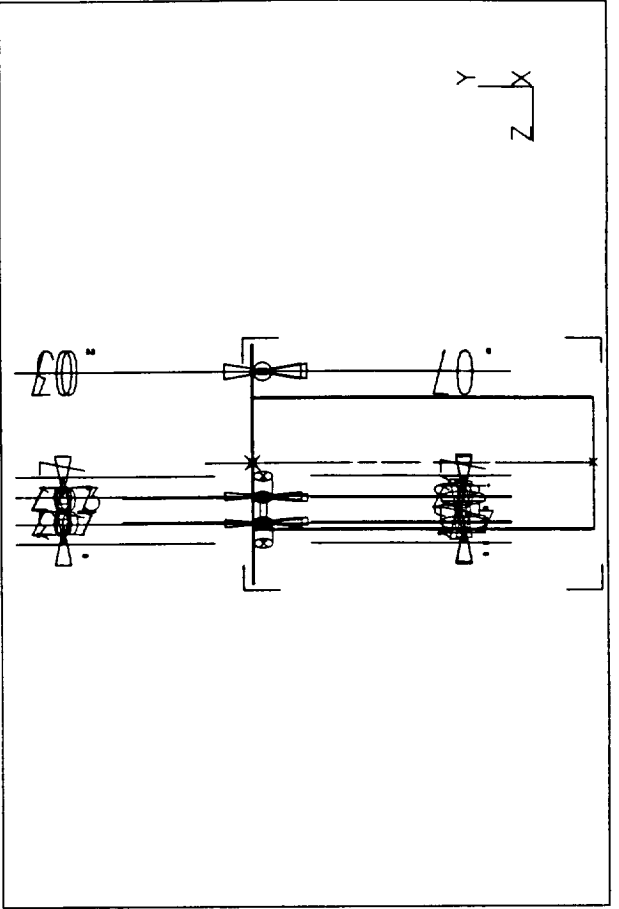
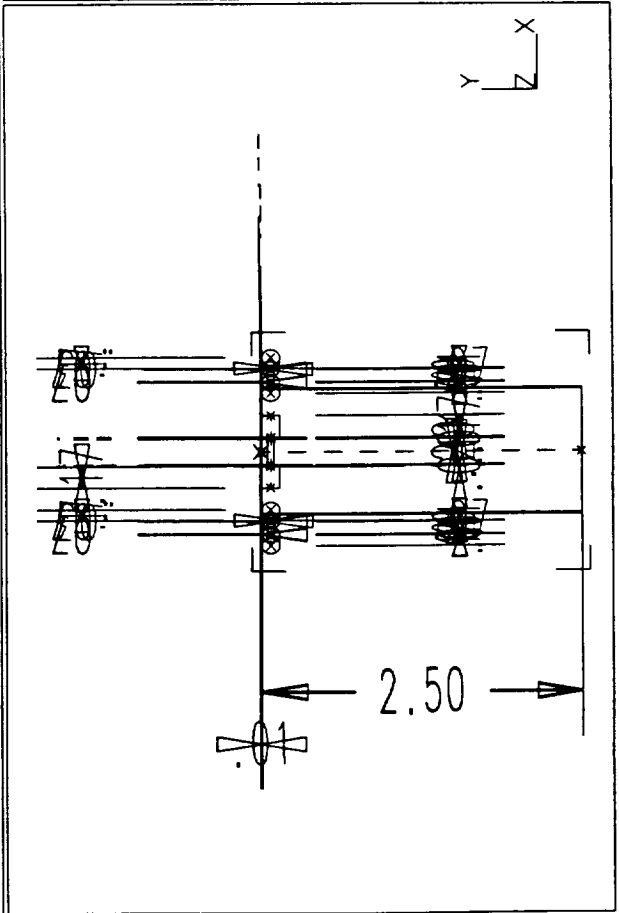
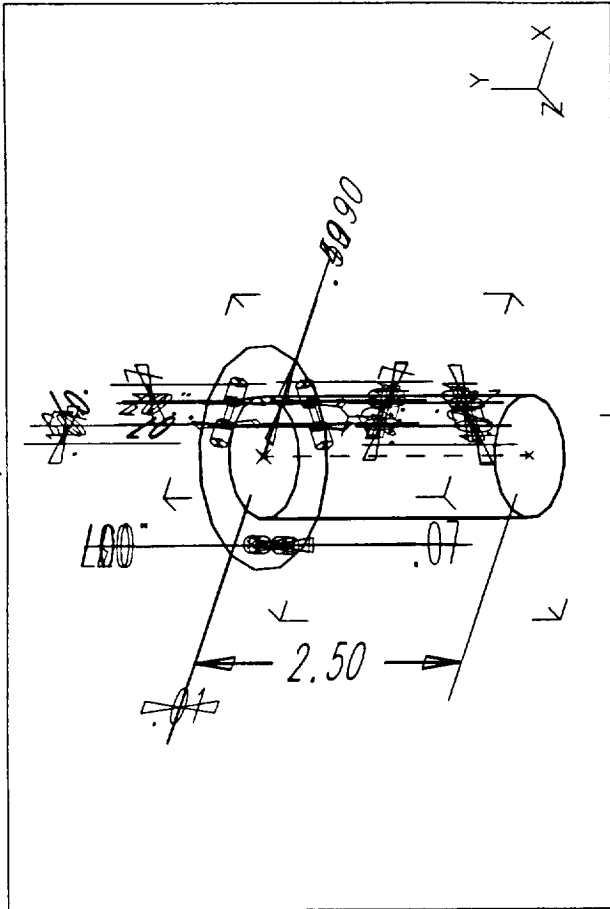
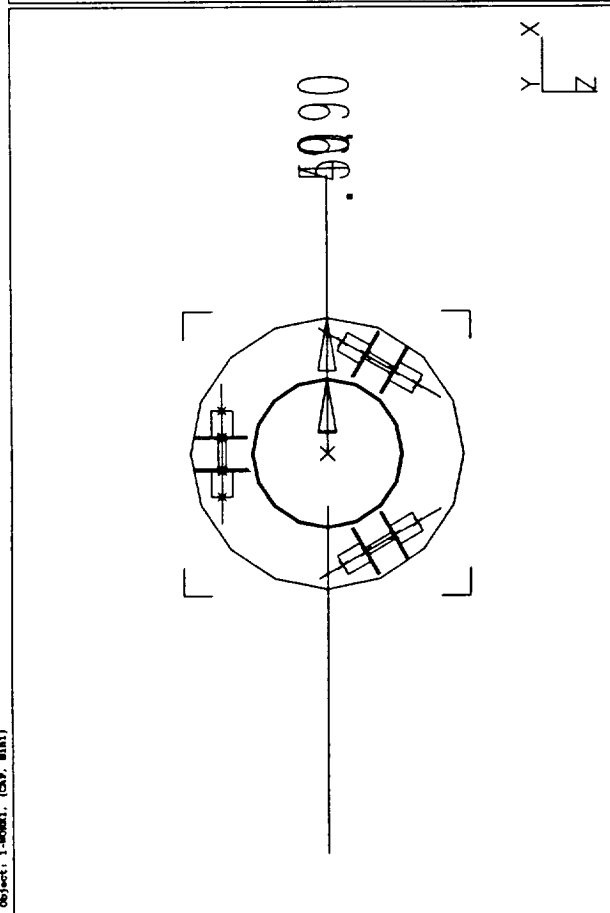
SDRC I-DEAS V11(6): Solid Modeling

15-MAR-94 14:51:08
Status : OK
Display : No stored Option
Update Level: Full
Size: 1-WIRE



5.6.A-11

Database MAY 1992
View : none, none, none, none
Task: Object
Object: 1-model1, (CSP, B1a1)



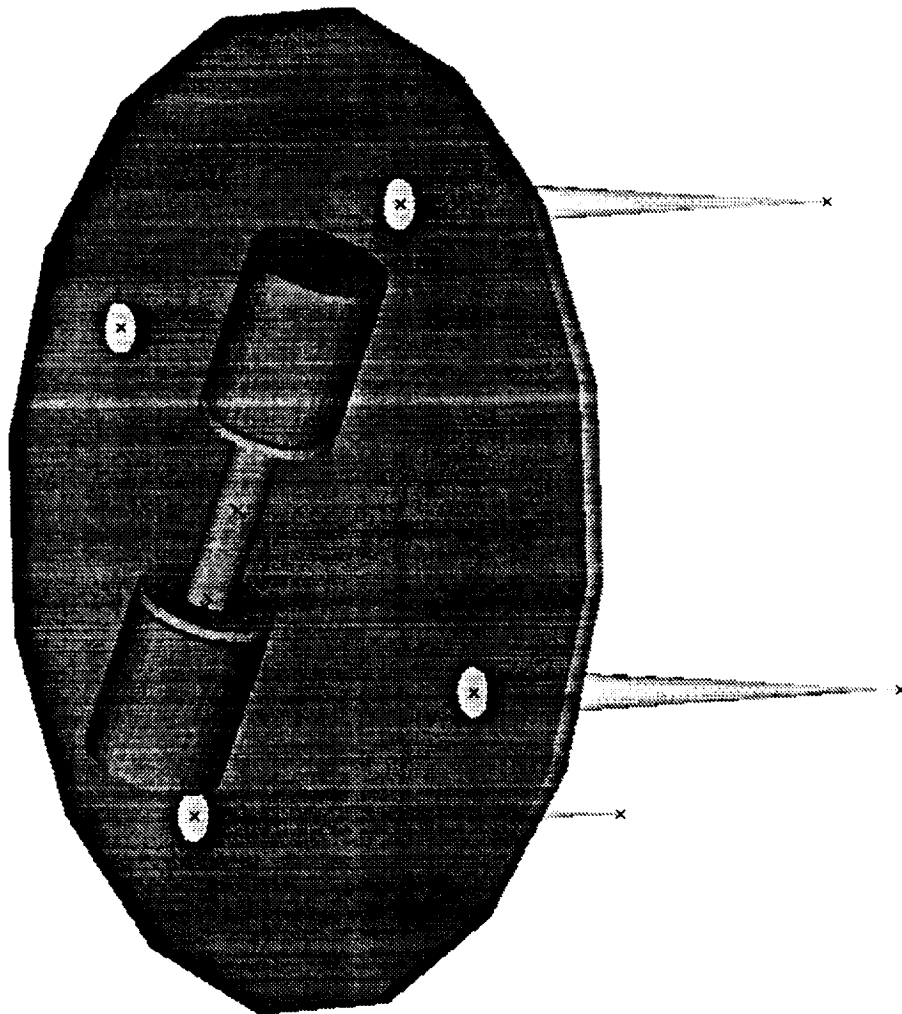
SDRC I-DEAS V6.i(s): Solid_Modeling

15-MAR-94 16:11:20

Units : #1
Display : No stored Option

Bin: 1-MAR94
Update Level: Full

Database: IRY_P012
View : No stored View
Task: Object
Object: 1-MAR94, (HATCH, B101)



15-MAR-94 16:10:41

Unit: in

Display: none, none, none, none

Bin: 1-MB18

Update Level: Full

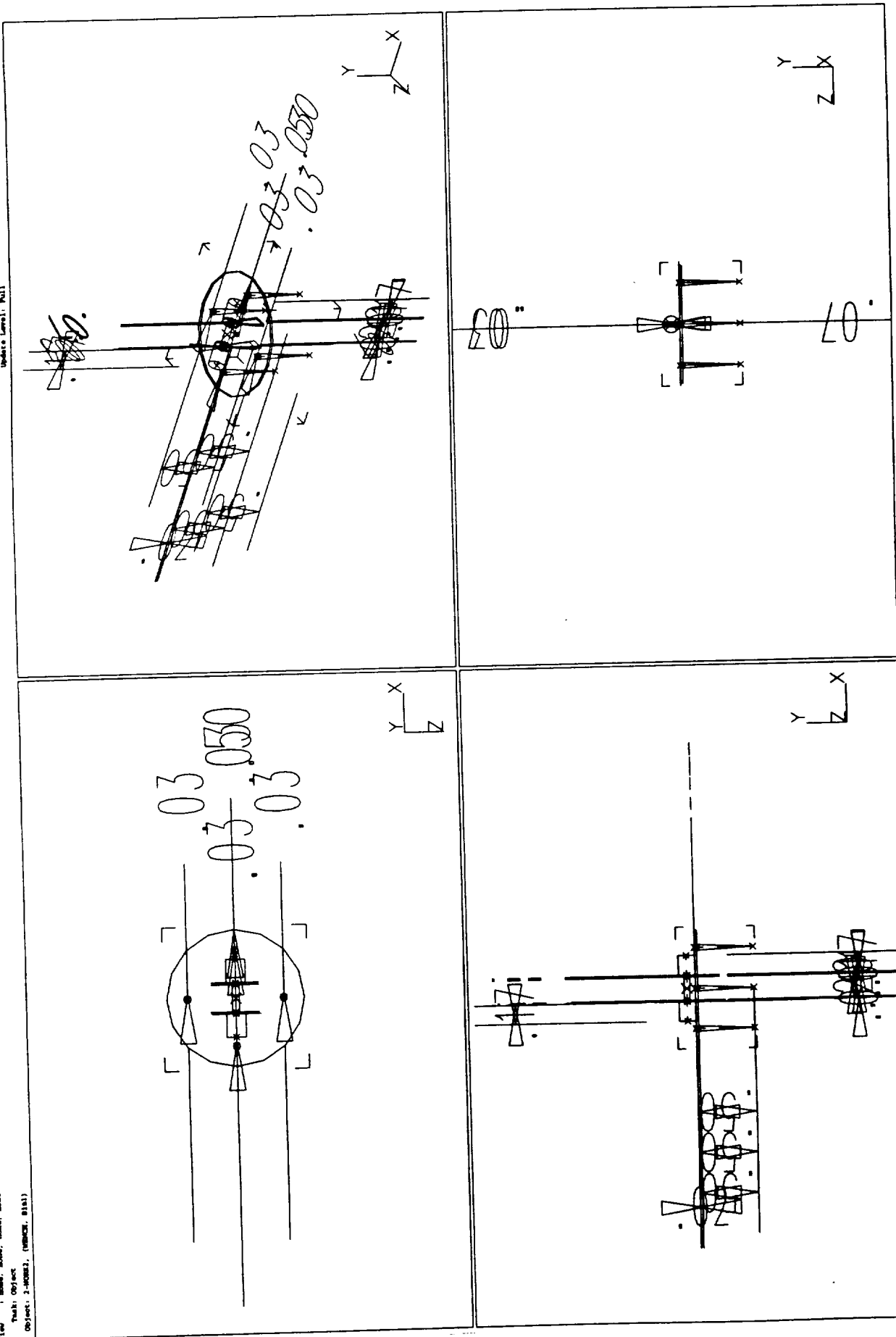
SDRC I-DEAS V6.1(s): Solid Modeling

Database: MKT_P018

View: none, none, none, none

Track: Object

Object: 3-MB181, (HATCH, B181)



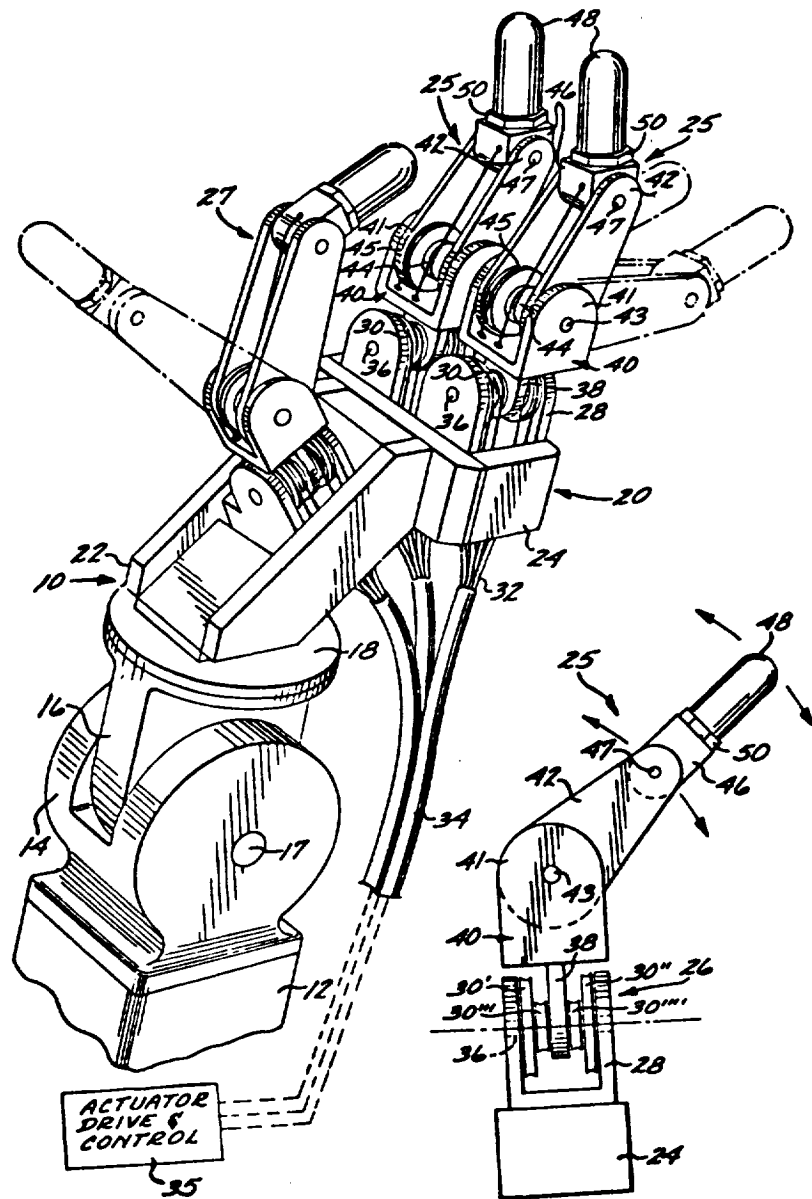


Figure 9-3. Stanford/JPL Hand. The hand is shown mounted on a Unimate 600 manipulator.

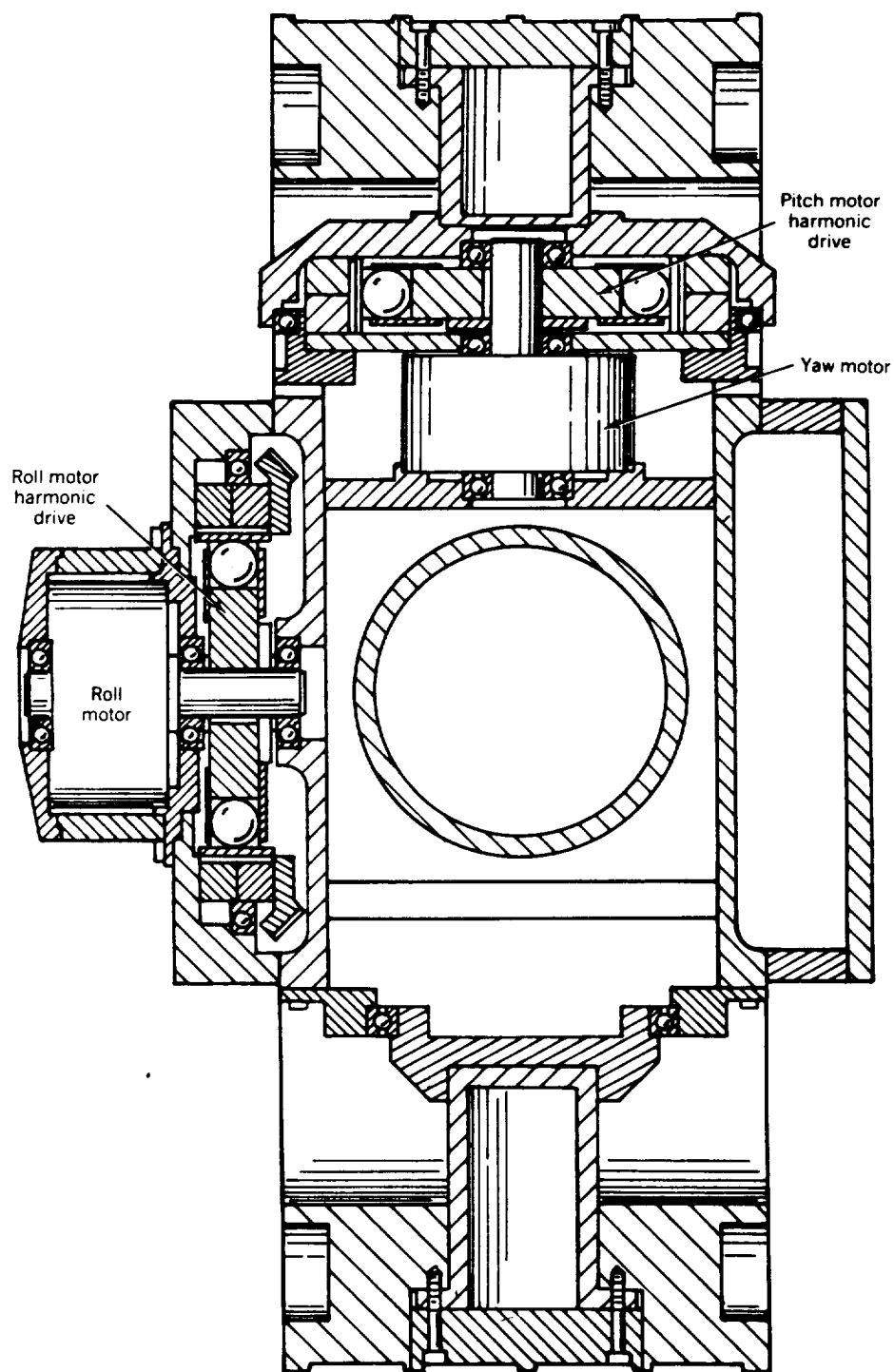


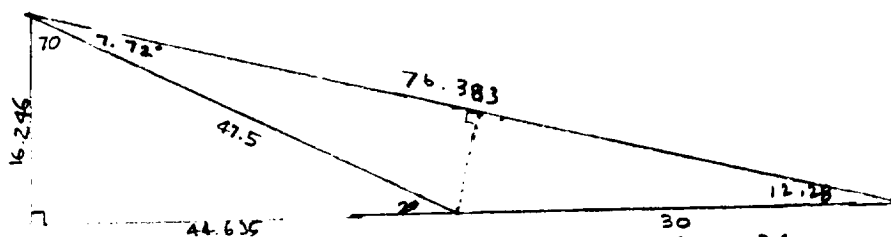
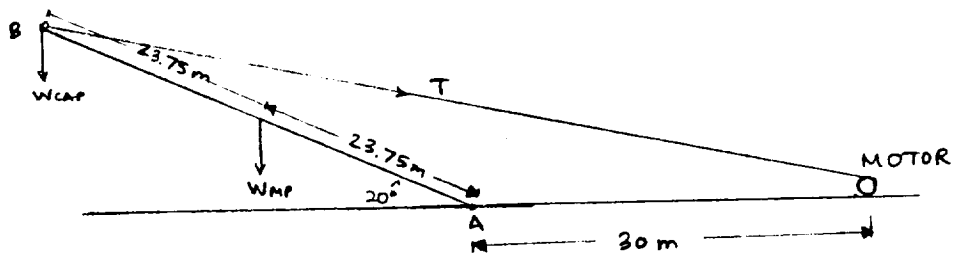
Figure 5.19. Sectional view of Powered Wrist Joint.

	Task	Time(sec)	Reps	per level	levels	TIME
	PREP of AREA					
1	Position Wench	300	6	1	1	1800
2	Obtain base platform	10	1	1	1	10
3	Lift base	15	1	1	1	15
4	transport to dome site	90	1	1	1	90
5	place on surface	20	1	1	1	20
6	obtain spike	10	3	1	1	30
7	screw in spike	45	3	1	1	135
8	move to next spike	10	3	1	1	30
9	move to staging area	45	1	1	1	45
	MAY POLE					2175
10	obtain maypole pieces cart	30	1	1	1	30
11	transport to dome site	90	1	1	1	90
12	turn to cart	5	5	1	1	25
13	grasp support	10	5	1	1	50
14	move to position	38	5	1	1	190
15	place support	10	5	1	1	50
16	move to cart	38	4	1	1	152
17	Turn to cart	5	1	1	1	5
18	grasp bottom piece	10	1	1	1	10
19	turn to site	10	1	1	1	10
20	place pole on pivot	30	1	1	1	30
21	turn to cart	5	1	1	1	5
22	grasp pin	10	1	1	1	10
23	turn to site	5	1	1	1	5
24	insert pin	20	1	1	1	20
25	lower pole	10	1	1	1	10
26	move to cart	18	5	1	1	90
27	grasp next pole	10	5	1	1	50
28	move to end of may pole	36	5	1	1	180
29	align pole	30	5	1	1	150
30	insert	15	5	1	1	75
31	grasp top pole	10	1	1	1	10
32	move to cart	30	1	1	1	30
33	grasp top ring	10	1	1	1	10
34	move to end of may pole	60	1	1	1	60
35	slide top ring on pole	10	1	1	1	10
	WIRES					1357
36	travel to stability wench	90	3	1	1	270
37	obtain cable	20	3	1	1	60
38	travel to pole	100	3	1	1	300
39	attach cable to pole	10	3	1	1	30
40	Travel to lifting wench	60	3	1	1	180
41	obtain lifting cable	20	3	1	1	60
42	travel to end of may pole	75	3	1	1	225
43	loop cable through pulleys	60	3	1	1	180
44	secure cable	15	3	1	1	45
45	attach to first ring	10	3	1	1	30
46	travel to staging area	45	1	1	1	45
47	obtain assistor	30	2	1	1	60
48	transport to 2/3 point of maypole	30	2	1	1	60
49	place on surface	10	2	1	1	20
50	slide under may pole	60	2	1	1	120
51	activate assistors	5	2	1	1	10
52	lift pole ~20 degrees	30	1	1	1	30

53	activate stability wenchers	5	1	1	1	5
54	lift pole to 90 degrees	120	1	1	1	120
55	secure wenchers	5	1	1	1	5
56	move to assistor	30	2	1	1	60
57	grasp	10	2	1	1	20
58	transport to staging area	45	2	1	1	90
59	drop	10	2	1	1	20
60	return to dome site	45	2	1	1	90
61	collect supports	38	5	1	1	190
62	place on cart	5	5	1	1	25
63	transport to staging area	90	1	1	1	90
64	obtain first level cart	10	1	1	1	10
65	transport to Dome area	90	1	1	1	90
CONSTRUCTION PROCESS						2540
66	turn to cart	5	2	7	9	630
67	grasp ring piece	10	2	7	9	1260
68	turn to site	5	2	7	9	630
69	connect to ring pieces	20	2	7	9	2520
70	move to next location	40	1	7	9	2520
71	turn to cart	5	4	7	9	1260
72	grasp pole	10	4	7	9	2520
73	turn to site	7	4	7	9	1764
74	lock top of pole	40	4	7	9	10080
75	lock bottom of pole	40	4	7	9	10080
76	grasp cart	10	1	7	9	630
77	move to next area	40	1	7	9	2520
78	release cart	5	1	7	9	315
79	grasp cart	10	1	1	9	90
80	transport to staging area	90	1	1	9	810
81	obtain next level cart	10	1	1	8	80
82	return to dome site	90	1	1	8	720
83	obtain camera	90	1	2	8	1440
84	mount camera	30	1	2	8	480
85	obtain platform cables	90	1	6	1	540
86	attach to structure	60	1	6	1	360
FINISHING						41249
87	set Dome on surface	30	1	1	1	30
88	obtain platform cables	30	6	1	1	180
89	attach to second pole from ground	10	6	1	1	60
90	lift slightly	20	1	1	1	20
91	remove pin	15	1	1	1	15
92	lift 1 meter	30	1	6	1	180
93	remove bottom pole	10	1	6	1	60
94	place on cart	20	1	6	1	120
95	lower to surface	60	1	6	1	360
96	remove platform cables	20	6	6	1	720
97	remove stability cables from top	20	3	1	1	60
98	retract stability cables	60	1	1	1	60
99	attach stewart platform to cables	30	1	6	1	180
100	transport pole to staging area	90	1	1	1	90
101	Pick up wenchers	235	6	1	1	1410
102	Remove material to staging area	90	1	1	1	90
						3635
						47411
TOTAL						71116.5
ML						19.75458
HRS						

8.6.C-2

LIFTING OF MAY POLE



$$\sum M_A = W_{mp} (23.75) (\cos 20^\circ) + W_{cap} (47.5) (\cos 20^\circ) - T (47.5) (\cos 7.72) = 0$$

$$T = 267.24 \text{ N}$$

$$A_{cross-sec} = \frac{T}{\sigma_{allow}} = \frac{267.24 \text{ N}}{1.75 \text{ GPa}} = 1.527 \times 10^{-7} \text{ m}^2$$

- USING $A_{cross-sec} = 1.963 \times 10^{-5} \text{ m}^2$ SO OKAY.

FOR KEVLAR CABLES.

- USE SAME MOTOR & DRUM USED FOR
DOME LIFTING

SDRC I-DEAS VI.1(s): Solid_Modeling

16-MAR-94 14:53:43

Unit : BG

Display : No stored Option

Bin: 1-MAIN

Update Level: Full

Database: jack

View : No stored View

Task: Object

Object: 9-WORK9, Workbench

